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TORQUE CONVERTER For Industrial and Commercial Vehicles

M. J. Waclawek, Borg & Beck Division, Borg-Warner Corp.

This paper was presented at a meeting of the SAE Chicago Section, Dec. 11, 1952.

ALTHOUGH in industrial applications of torque converters the problems are usually much different from the automotive ones with which we have the most experience, many of the old advantages are still realized and a number of new advantages become apparent.

The first advantage, and probably the prime purpose of a torque converter, is the continually variable torque ratio obtainable. This gives a smooth acceleration of the load up to speed without the inconvenience and time loss entailed in shifting to obtain the desired ratios. The low drag characteristic of a torque converter at idle speeds is advantageous in that the engine need not be de-

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TORQUE converters have many advantages for industrial and commercial vehicles. Some of those listed by the author are:

1. Smooth acceleration.

2. Engine need not be declutched when it is unloaded or idling.

3. More horsepower delivered at low output speeds and during acceleration, compared with conventional transmissions.

4. Operator mistakes of improper gear selection and missing of shifts are reduced to a minimum.

5. Load held to power source at all times.

6. Damping out of engine torsional vibration.

7. Increased life for drive train beyond transmission.

8. Cushioning of shock loads during starting and shifting, and of sudden increases in load.

The Author

M. J. WACLAWEK (J '49) has been project engineer at the Borg & Beck Division of Borg-Warner Corp. since 1951. He received his B.S. degree from Worcester Polytechnic Institute and his M.S. in automotive engineering from the Chrysler Graduate School of Engineering. After serving in the Navy as an officer in the Supply Corps Mr. Waclawek spent four years at Chrysler and also was a project engineer for Bell Aircraft in 1950.

clutched when it is unloaded or idling.

Because the stall speed of a torque converter transmission is usually high, more engine horsepower is delivered at low output speeds and during the acceleration period when the load is coming up to speed.

By eliminating or minimizing gear shifting, operator mistakes of improper gear selection and missing of shifts when necessary are reduced to a minimum. Here we also have the advantage of the torque converter holding the load to the power source at all times.

In applications such as hoists, winches, and cranes, where it is sometimes desirable to hold a load suspended for any length of time, the torque converter can perform this without racing the engine and slipping a clutch.

The next advantage is the damping out of engine torsional vibrations, which is due to the cushioning effect of the fluid. This eliminates the stress variations caused by torsionals and resonance. Consequently, it is possible to employ higher design stresses in transmission parts, feasible to use higher loading, or to obtain longer life for equal loading. In automotive applications of torque converters, we have experienced less wear in gears and splines and have been able to increase the capacity of the transmission. An example is the torque converter output shaft in the Merc-o-matic which is 27/32 in. in diameter and carries over twice engine torque, while the transmission input shaft in the Mercury standard transmission carries engine torque only and is 1 in. in diameter.

A remarkable increase in the life of the drive train beyond the transmission has also been reported.

Another advantage of the torque converter drive is the cushioning of shock loads during starting and shifting. Probably even more important is the cushioning of sudden increases in load. The torque converter has the added advantage here in that the increased load will automatically increase the torque ratio and, in the extreme case, with the output completely stopped, will give full stall ratio without stalling the engine.

Another advantage of our torque converter, which is most important to the industrial user, is its simple construction. If servicing is ever required, the unit can be disassembled and repaired quickly and economically.

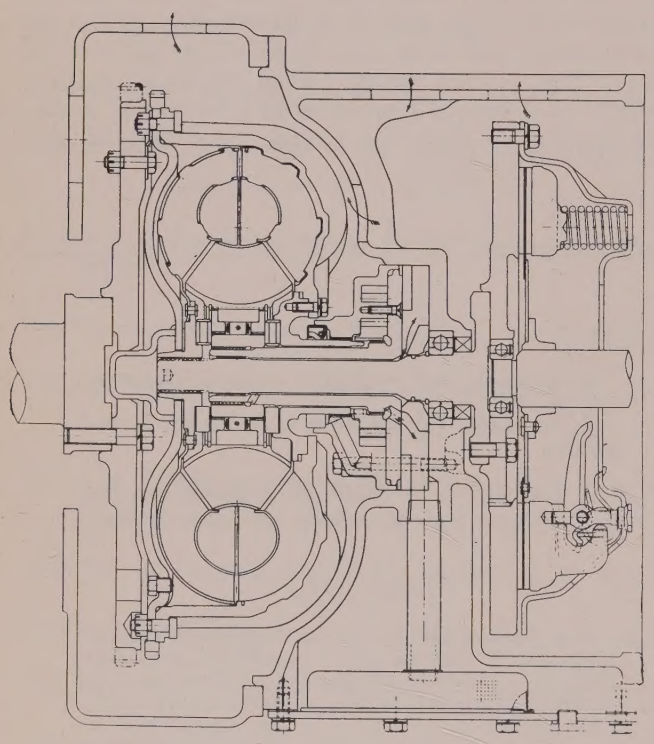

Fig. 1 – Borg & Beck torque converter and clutch assembly with Warner Gear oil pump in typical housing recommended for industrial use

Construction and Servicing

Before torque converter applications are discussed, a brief description of a torque converter adopted for industrial use should be enlightening.

Fig. 1 is a cross-section of a Borg & Beck torque converter and clutch assembly with a Warner Gear oil pump in a typical housing recommended for industrial use. The torque converter and oil pump are those now used in the Ford-o-matic and Merc-o-matic transmissions as shown in Fig. 2.

This torque converter is a 3-element, regenerative type in which the fluid enters the impeller directly from the reaction stage. In this arrangement, the fluid leaving the reaction stage has a forward direction, so that the impeller has less work to do on it. This results in the highest efficiency in both torque converter and coupling range and still gives a reasonably high stall torque ratio.

Smooth, accurate blade contours are produced by die castings and cold fabrication of steel blades, instead of the more common sand and plaster castings or brazed and welded stampings.

The stator and the impeller housing are both aluminum die castings. The outside of the impeller housing has 68 cooling fins and the inside has four rows of slots for 31 blades. The blades fit into the slots in the housing and are anchored in place by a snap-ring at one end and rolled-down tabs on the inner shroud. The stator is die cast as a unit having 11 airfoil blades integral with the hub and shrouded with a fabricated steel band. This gives all the advantages of an airfoil without the roughness and misalignment of cores associated with sand or plaster castings.

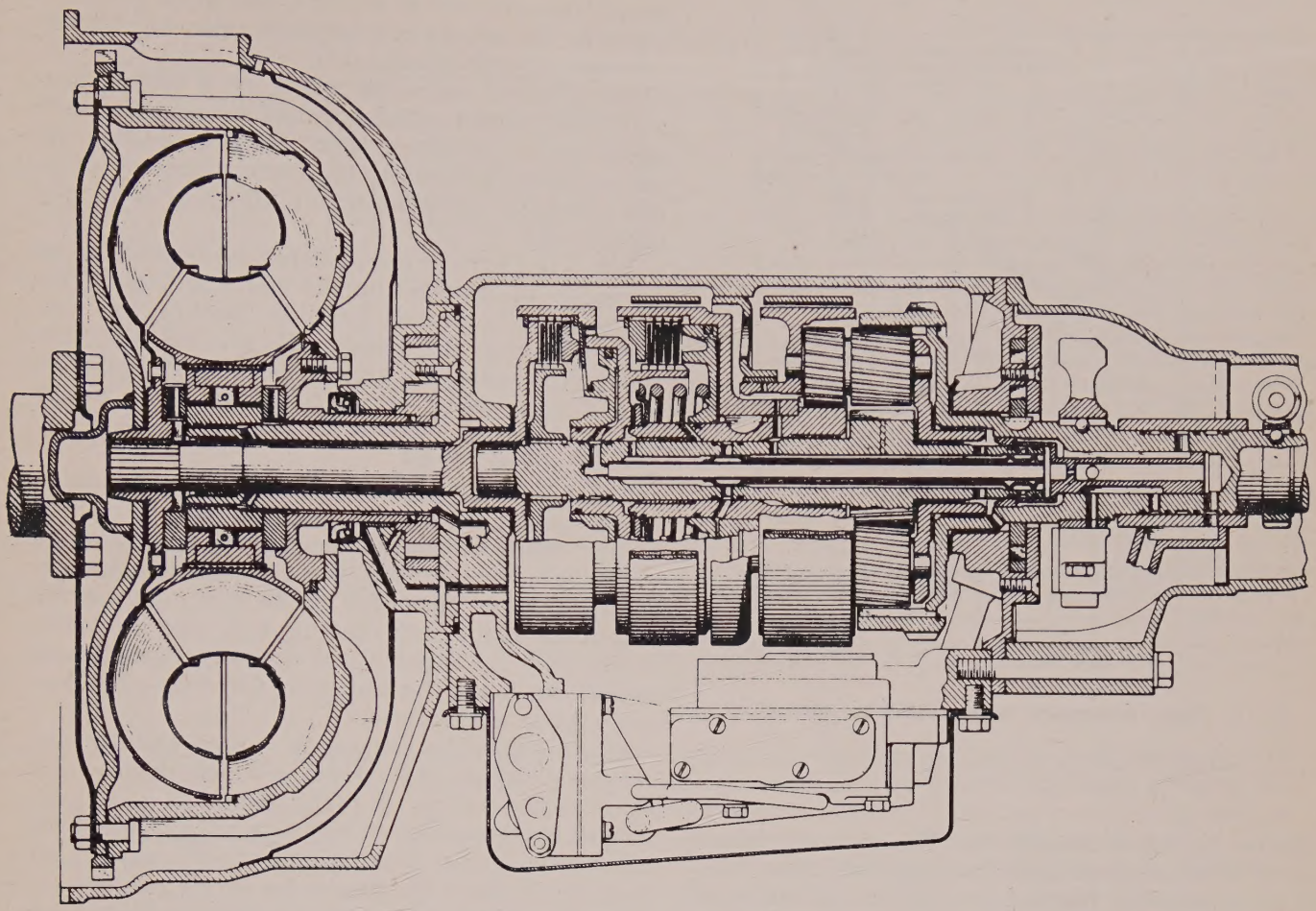

Fig. 2 – Torque converter and oil pump used in Ford-o-matic and Merc-o-matic transmissions

The turbine blades are made with six tabs each that fit into slots in the inner and outer shrouds. The tabs are then rolled down, forming a sturdy blade assembly, which positions the blades very accurately and yet is capable of damping out any noises that tend to originate in the turbine.

The turbine is splined to the torque converter output shaft, which is, in turn, piloted by a bushing in the stator support shaft. The front or engine end of the converter is piloted in the crankshaft end or an adapter at the crankshaft, while the rear hub is in a bushing between the impeller hub and transmission case. The result of this arrangement is that the torque converter has a chance to breathe when fluid pressure increases, and also any axial misalignment will adjust itself without inducing bearing-fight or undue stress in the shafts.

The stator is mounted on a stationary shaft through a one-way clutch so it can rotate in the forward direction only. The lockup sprags in the one-way clutch are mounted in a cage so as to give a self-contained unit that can be assembled very quickly. This one-way clutch is gaining popularity because of its simplicity and reliability.

The impeller hub drives the oil pump, which charges the converter to a pressure of 50 psi and circulates the oil at about 2 gpm at 800 rpm of the engine. This pump could also be used to supply a limited amount of oil to perform some other outside functions, if necessary.

Characteristics

The most important information for the designer and prospective user of torque converters are the characteristics to be expected. This is where torque converters vary considerably from one another.

Because the torque converter is a hydraulic machine, its characteristics will be the same under

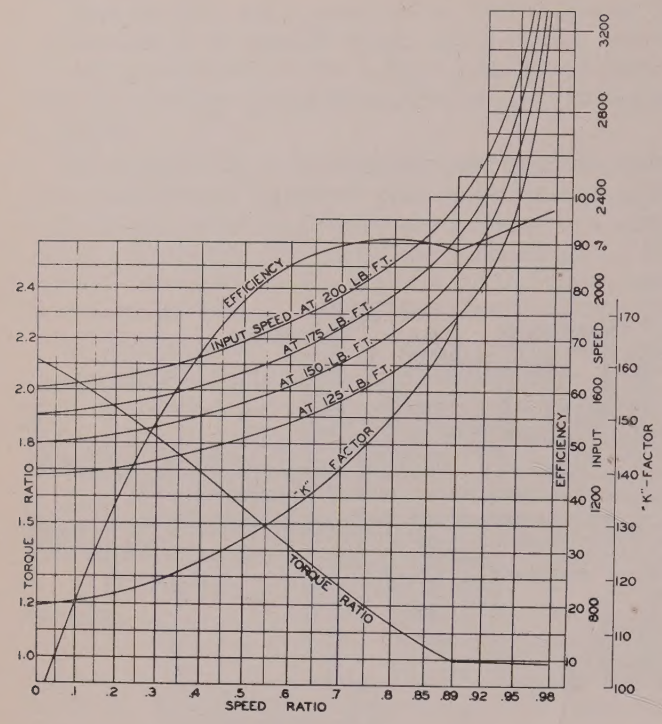

Fig. 3 – Typical performance curve supplied by manufacturer

similar fluid-flow conditions. The ratio of output speed to input speed determines the fluid-flow directions between the blades; thus, the torque ratio and efficiency will be constant at each speed ratio, regardless of the torque input. The variation of the torque input will affect the magnitude of the fluid flow, which will in turn vary the input speed. Because torque is a function of the square of the speed, the quantity K which is the input speed divided by the square root of the torque, will also be constant at each speed ratio.

Each torque converter will have a certain constant torque ratio, efficiency, and K factor for each speed ratio. Knowing these three quantities, we can calculate accurately the performance of a torque converter installation for all loads and input conditions.

Fig. 3 is a typical performance curve of a torque converter as supplied by the manufacturer. This is the form preferred by those working on converters because the torque ratio and efficiency are always the same at a given speed ratio, regardless of the input speed or input torque. One torque ratio and one efficiency curve serve a whole family of input speed curves.

Interpolation between the curves shown allows an input speed curve to be plotted for odd torque values. Unfortunately, since engine torque varies with engine speed, this is not a particularly convenient form of curve for engine men. To get the output speed for each reading, it is also necessary to multiply the input speed times the speed ratio so that this is not the best curve for vehicle builders and application engineers.

Fig. 4 shows the same data plotted against output speed instead of speed ratio, which gives a more direct visualization of performance in a vehicle. Output speed can be converted to vehicle speed for any particular application and can be marked off along the base of the graph for direct reading.

On this curve, it is readily apparent that torque ratio increases with input torque at a given output or vehicle speed, which is the chief advantage of a

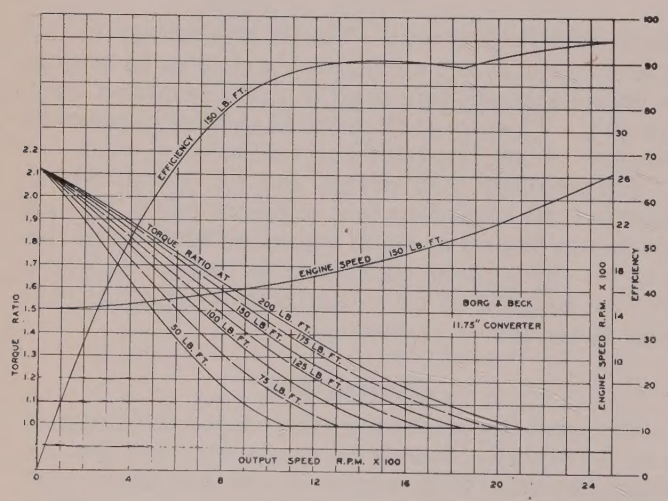

Fig. 4 – Data plotted against output speed

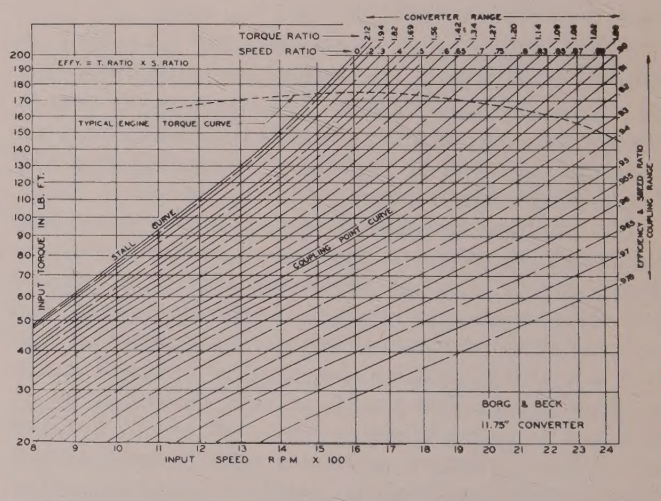

Fig. 5 – Most convenient method of plotting data for vehicle builders and application engineers

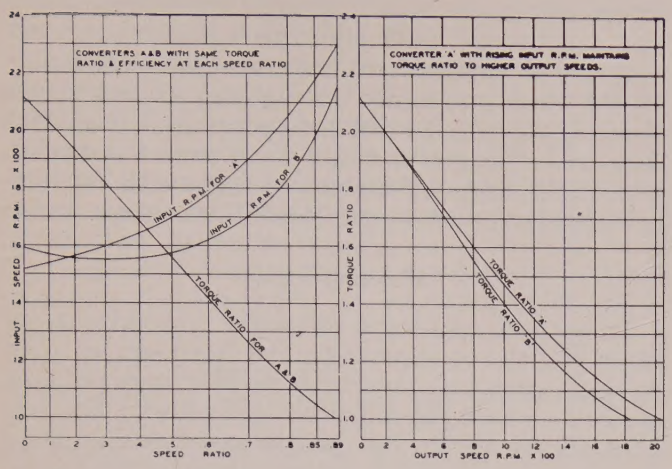
CONVERTERS A & B WITH SAME TORQUE RATIO & EFFICIENCY AT EACH SPEED RATIO
CONVERTER 'A' WITH RISING INPUT R.P.M. MAINTAINS TORQUE RATIO TO HIGHER OUTPUT SPEEDS.
INPUT SPEED R.P.M. X 100
INPUT R.P.M. FOR A
INPUT R.P.M. FOR B
TORQUE RATIO FOR A & B
TORQUE RATIO
SPEED RATIO
TORQUE RATIO A
TORQUE RATIO B
OUTPUT SPEED R.P.M. X 100

Fig. 6 – Speed ratio and output speed curves for two converters having similar torque ratio characteristics but widely varying input speed characteristics

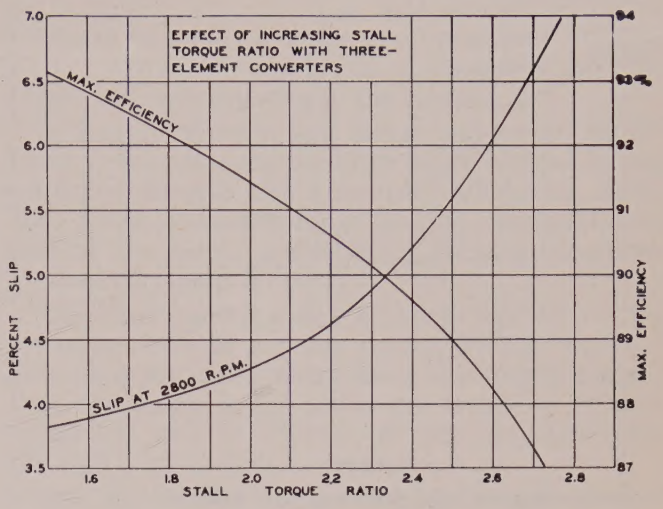

Fig. 7 – Efficiencies obtainable in present stage of development of 3-element torque converters for various stall ratio converters

converter in giving flexible performance under varying load conditions. For example, at 1000 rpm output, the torque ratio can be varied from 1 to 1 at 45 ft-lb engine torque to 1.50 to 1 at 200 ft-lb torque, giving an increase in output from 45 ft-lb up to 300 ft-lb, by increasing the throttle at this speed. Furthermore, if an increased load should slow the output down to, say, 400 rpm, the torque ratio will further increase to 1.89 to 1, giving 378 ft-lb output torque.

The most convenient representation of torque converter characteristics for vehicle builders and application engineers is shown in Fig. 5. By plotting curves of constant torque ratio and speed ratio on coordinates of input torque versus input speed, we can draw in any engine torque curve on the same coordinates to exact values without interpolation. Intersections of the engine torque curve with the performance curves (followed diagonally to the top scale) give torque ratio and speed ratio. Intersections followed down a vertical line to the bottom scale give the input speed. Output torque, output speed, and efficiency are then obtained by multiplying the values indicated on the chart. The effect of a governor can also be shown on this chart.

Fig. 6 shows speed ratio and output speed curves of two converters having similar torque ratio characteristics but widely varying input speed characteristics. These curves illustrate the importance of not only the torque ratio but also of the K factor when a converter application is considered.

From the curves, it is apparent that, even though the torque ratio is the same at a specific speed ratio, it varies considerably when plotted against output speed. The reason for this is that the higher engine speed at any speed ratio will have a higher output speed, and the difference in output speeds becomes larger as the speed ratio increases.

The right-hand curve of Fig. 6 shows that at an output speed of 1830 rpm, converter B is at 1 to 1

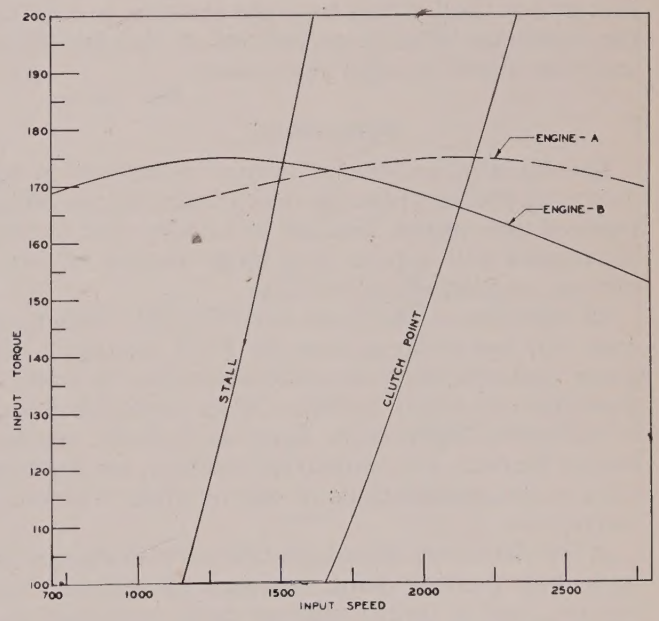

Fig. 8 – Curves for high-speed engine A and low-speed engine B

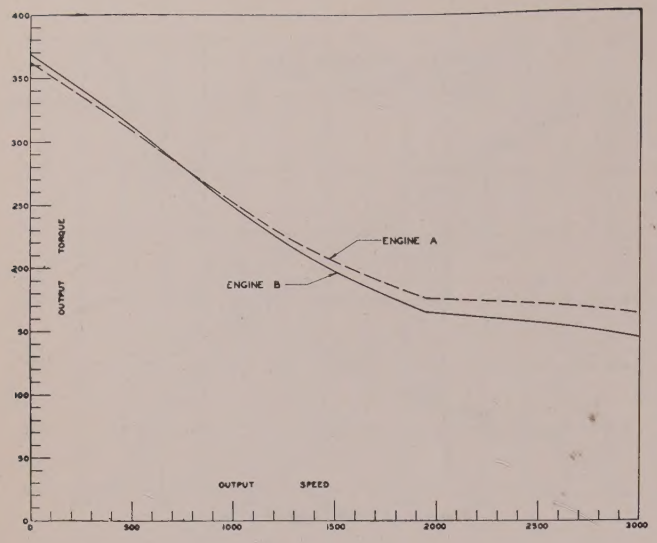

Fig. 9 – Output torque versus output speed of converter with engines A and B

torque ratio, while converter A still has 1.07 multiplication. Part of this gain can be lost, however, if a low-speed engine is used, as will be demonstrated later.

Fig. 7 shows the efficiencies obtainable in the present stage of development of 3-element torque converters for various stall ratio converters. This curve graphically illustrates the sacrifices that must be made in both the coupling efficiency and maximum converter efficiency if larger stall ratios are desired. In most cases where high ratio converters are used, it has been the practice to lock up the converter when it comes out of multiplication and thus avoid the high slip losses.

Applications

Specific applications of torque converters in industry at present, plus the many installations being planned and tested, lead us to believe that torque converters will appear in a large portion of commercial and industrial vehicles.

In addition to the Ford and Mercury passenger cars, our converter is used on Ford industrial engines, and two truck manufacturers plan to make it available on certain models. Also, several builders of industrial equipment, such as fork-lift trucks, bucket loaders, and industrial tractors, are making pilot model installations of our or other 3-element converters.

A few farm and industrial tractor manufacturers have been testing pilot models with torque converters, and a larger number have expressed interest in starting a test program in the near future. City buses have been using torque converters for a number of years and demonstrate their desirability for stop-and-go service.

Other converter applications being planned or in effect are cranes, hoists, shovels, switchers, crawlers, and trucks. Although the economies of highway truck applications have not yet been established, there are many off-highway trucks in use showing increased work done per unit of time with less driver fatigue and less down time for service. Over 1200 large, off-highway trucks with converters are now in use in the Mesabi iron range alone.

Choosing Engine

In working on torque converter applications, we often find the equipment builder handicapped with an engine having characteristics that were thought desirable before the converter was added. The lugging ability shown by the low-speed engine B in Fig. 8 has been highly valued because its torque increases when more load is encountered, resulting in less tendency to stall out and less need for downshifting. With a torque converter in the line, this engine cannot stall out and cannot even be loaded down to the speed of its best torque.

The high-speed engine A sacrifices some torque at stall, but has its best torque within the operating range and delivers more horsepower for the same displacement.

Fig. 9 shows the output torque versus output speed of the converter with engines A and B of Fig. 8. The high-speed engine shows a considerable advantage because the gain in torque occurs in a speed range where the engine will do most of its operating.

Fig. 10 shows the importance of choosing a torque converter of the correct capacity. Here we have engines of three different sizes used with the same converter. We will see what performance each will give. It should be noticed that each engine has its own governor speed, the biggest engine having the lowest speed and the smallest having the highest speed. Although governor speeds vary considerably among engine manufacturers, each manufacturer generally has the largest engines governed at the lowest speeds.

In Fig. 10 the curve marked "stall point" denotes the maximum torque ratio and occurs when the output speed is zero. The curve marked "clutch point" is the point where the torque ratio becomes one and the torque converter starts acting as a fluid coupling. Between the two curves there is torque multiplication, starting with maximum ratio at stall and gradually diminishing until it becomes one at the clutch point.

Engine A is too large for this converter because (1) its maximum torque will never be utilized as it occurs at a lower speed than the stall speed of the converter and (2) the governor starts diminishing engine torque while the torque converter is still multiplying, which means the converter input torque will be less than what the engine is capable of supplying at that speed.

Engine B is near the optimum size for this converter. It should be noted that the engine torque peaks at stall and the governor starts acting beyond the clutch point, well in the coupling range.

Engine C is probably too small for this torque converter because it has too low a stall speed for most applications and comes out of torque multiplication too soon. On the curve, it is seen that multiplication occurs from 1000 to 1500 rpm engine speed, and from 1500 to 2400 rpm the output torque is equal to the engine torque.

These same three engines and the same torque converter appear in Fig. 11, plotted against output characteristics. Engine A is shown without a torque converter, and with a torque converter that is governed at 1800 rpm, and not governed at all. It is evident that the torque converter without the governor gives the best performance. When a governor is added and it starts acting in the torque conversion range as shown, a large portion of the torque converter potential is lost. In fact, from an output speed of 1250 rpm to the governed speed of the engine, engine A without a torque converter gives decidedly more output torque than with a converter when governed.

Engine B on the output curve gives an optimum utilization of a torque converter for best performance. Now let us see what happens if we use engine B with the same load we intended for engine A. In order to get the same output speed for B as we got for A, we will have to change the rear axle or the gearbox ratio by 1.168, because the two engines are governed at different speeds. This means that the torque of engine B, as it goes through the gears, will increase 16.8% more than that of engine A. In the operating range, engine B will be running faster because its governor setting is higher. Thus, the operating efficiency of engine B will be higher because the torque converter is in the coupling range at this point, and coupling efficiency increases with engine speed.

It is seen that the much smaller engine B will almost equal the stall performance of engine A and surpass it in the operating range.

Engine C is considered too small for the torque converter shown in that it gets out of ratio too soon. Although the coupling efficiency is very good at the governed speed, it is usually desirable to have torque multiplication over a larger range of output speeds.

Since there is a need for engines of higher speed with torque converters, and since there are more industrial engines commercially available than there are torque converters, it becomes doubly important that the correct engine be selected in order to realize the biggest advantages of a torque converter application.

Oil Supply

A torque converter is a hydraulic machine and as such needs a working fluid. Oil is almost universally used in torque converters because of its properties, cost, and its service as a lubricant. For best performance, the working fluid should be under pressure to avoid cavitation and provide a circulation. Cavitation causes loss of efficiency, is very noisy, and is destructive to the blades.

Although there is one torque converter manufacturer who is using the differential of head in the converter to give him a natural circulation, most have made provisions for a positive charging pressure by one means or another. Charging pressures are usually about 50 psi.

The most common way of supplying oil to the converter is with a pump driven by the impeller hub. In some cases where there is a length limitation, the pump is set below the torque converter centerline and is driven off the impeller hub through a chain drive or gears.

Another method of charging the torque converter is one that was just recently announced by Chrysler, and that is by using engine oil pressure. This seems like a very good idea if a detergent oil and good filters are used to avoid depositing dirt and sludge in the converter by centrifuge action.

Many industrial vehicles already have high-capacity oil pumps for operating auxiliary equipment. Oil supplied by this hydraulic system could be used to pressure charge the torque converter too.

General Motors diesel-driven buses at one time

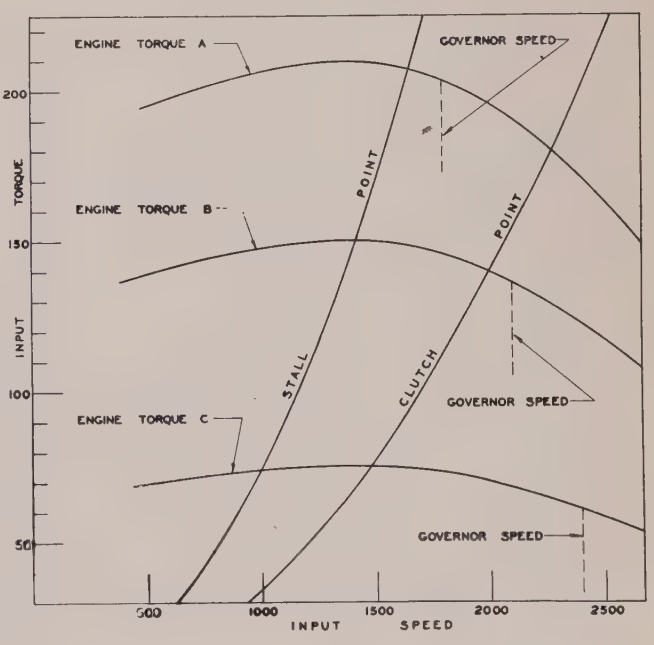

Fig. 10 – Input torque versus input speed for engines of three different sizes

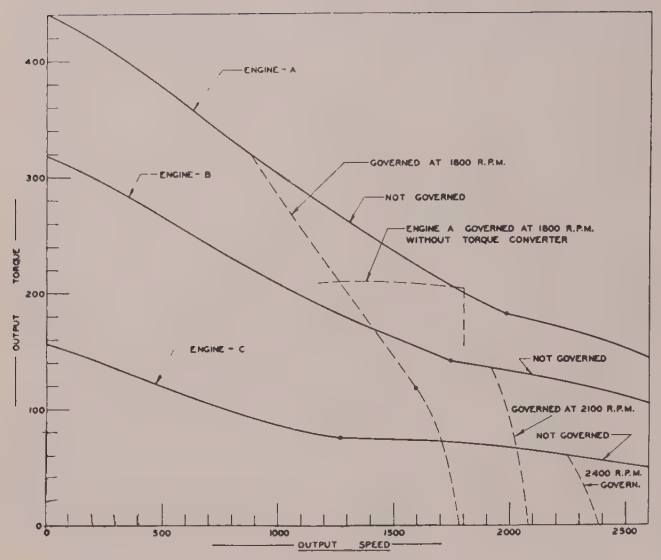

Fig. 11 – Output torque versus output speed of same converter with same three engines as shown in Fig. 10

were charging their torque converters with diesel fuel pumped directly from the fuel tank.

The best means for fluid circulation in an industrial installation will vary with application. Although some engine manufacturers frown on the idea of using engine oil to charge a converter, we have found that space limitation is a problem with many industrial equipment builders and they have no alternative but to eliminate the pump.

The problem of oil cooling will probably be more prevalent in industrial applications than in automotive. Our torque converter was designed with 68 aluminum fins on the outside of the impeller, which would circulate outside air around the converter and dissipate the heat encountered during all normal driving. It is reasonable to expect that there are many industrial applications where this converter would not get sufficient cooling unless other means were provided. Fig. 12 is the heat-dissipating capacity curve of our 11¾ in. torque converter.

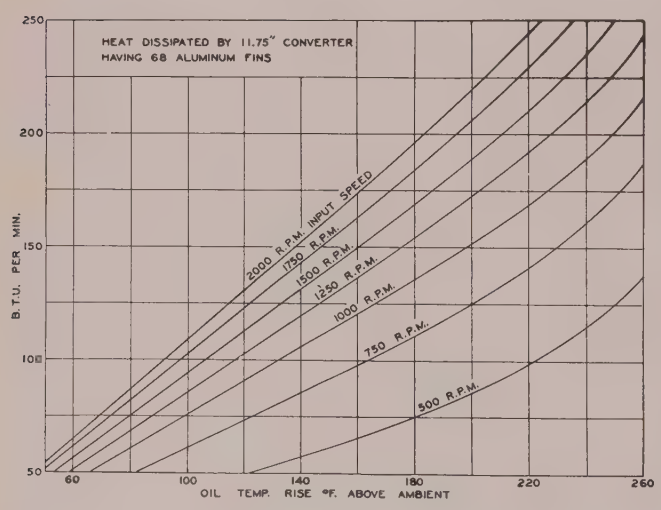

Fig. 12 – Heat-dissipating capacity curve of Borg & Beck 11¾-in torque converter

An outside heat exchanger connected to the engine cooling water is the method most widely used at present to keep the oil temperature of the torque converter down.

Clutches

Shifting a sliding gearbox behind a converter with the vehicle at rest can be accomplished with a conventional clutch, a rocking turbine brake, or a hydraulically applied clutch. The turbine brake, however, gives trouble if the vehicle is moving or is on a grade. Also, an automatic hydraulic clutch does not meet the requirements of lift trucks, farm tractors, earthmovers, and bucket loaders, where the engine speed is manipulated independently to control the auxiliary equipment while the clutch is slipped deliberately to maneuver the vehicle. The conventional friction clutch, as shown in Fig. 1, seems to be the most versatile and economical solution, and in spite of the increased torque transmitted by the converter we find it can usually be made smaller than the same application required without the converter. This has been done by designing the clutch to accommodate about twice the usual spring pressure. An example of this is our 9½-in. clutch used behind the DeSoto V8 engine and converter having nearly three times the stall torque of the DeSoto 6-cyl engine using the 10-in. clutch without a converter.

Flywheel Requirements

The Borg & Beck and other automotive converters are furnished with a starting ring gear and are used in passenger cars without an engine flywheel. But their polar inertia is not adequate for some 6- and most 4-cyl industrial engines, and an auxiliary flywheel, as shown in Fig. 1, is recommended to give better idling. This also provides a good mounting arrangement using tangential steel straps between converter and flywheel, and also permits other starting gear sizes to be mounted when desired, as shown in this figure. Sometimes an internal-external gear or spline mounting is used to eliminate the need for access holes in the bell housing for bolting a drive strap or flex plate; but these cost more, have noise problems, and require endwise constraint of the converter.

Acknowledgment

The author would like to express his sincere thanks to Richard Smirl, assistant chief engineer and assistant director of engineering at the Borg & Beck Division of Borg-Warner Corp., for not only suggesting the presentation of this paper, but also for the time he spent in helping and the many ideas he contributed to the text itself.

THE POSTWAR CAR AND THE A-BLAST . . . WHAT ARE THE ODDS?

REPORT OF THE

SAE-Federal Civil Defense Administration Advisory Committee

Presented by A. L. HAYNES, chairman, at the SAE Summer Meeting, Atlantic City, June 9, 1953.

IT was in December, 1952, that the Federal Civil Defense Administration invited the SAE to assist in planning, conducting, and evaluating the technical aspects of the effects of an atomic explosion on a number of typical American automobiles.

The purpose of the test was twofold:

1. To determine the amount of protection against blast, heat, and radiation that would be afforded to people in cars at the time of an atomic attack.

2. To determine the effects of blast, heat, and radiation on the vehicles themselves, particularly in regard to their usefulness after an atomic attack.

The SAE was primarily concerned with the latter objective.

It was the desire of FCDA to obtain data, opin-

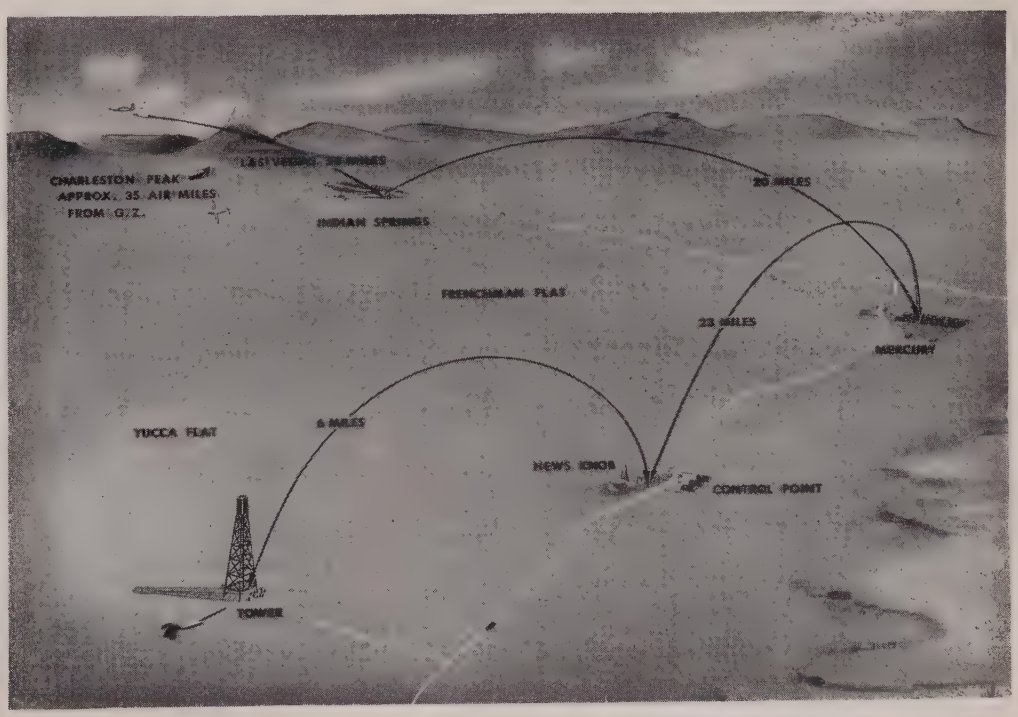

Fig. 1 – Map showing area in which atomic test took place

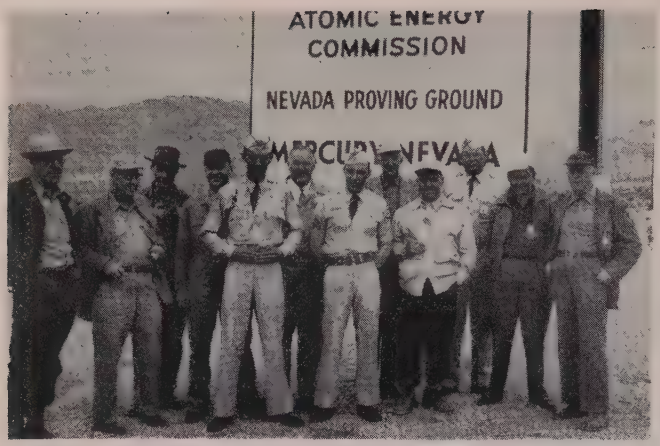

Fig. 2 – Members of SAE-FCDA Advisory Committee with AEC security officers at entrance to Nevada proving ground Mercury, Nev. (left to right): Dr. E. J. Martin, N. F. Hadley, R. B. Alexander, Dr. V. B. Corey, AEC officer, M. J. Muzzy, AEC officer, W. W. Smith, Chairman A. L. Haynes, AEC officer, W. Christensen, and R. C. Sackett

Fig. 3 – Test cars lined up in Helldorado Park, where SAE-FCDA Advisory committee members made pretest evaluation

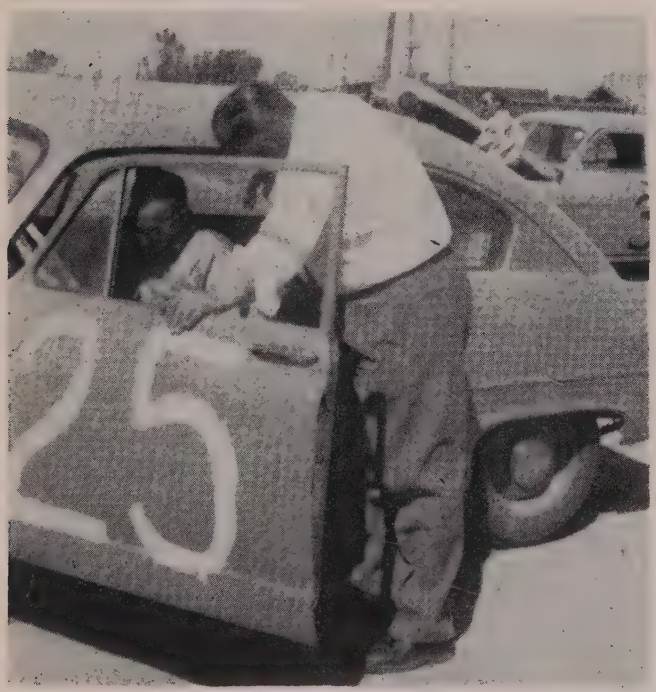

Fig. 4 – Car being checked before test by N. F. Hadley, (seated) and M. J. Muzzy, (standing)

Fig. 5 – A. L. Haynes (left) and V. B. Corey helping in pretest check

Fig. 6 – W. W. Smith (left) and W. Christensen making pretest check of car

Fig. 7 – N. F. Hadley and M. J. Muzzy making pretest check of car

WHAT would happen if an atomic bomb were to explode at noon tomorrow without warning of any kind? Would the automobile turn out to be a death trap for many of us or would it offer as much protection as the ordinary building? Would there be any advantage in having the car turned away from the center of the blast or to having the windows rolled down? Would it be safer to crouch down on the floor of the car at the first sign of an atomic explosion? Would it be safe to enter our cars as soon as the first effects of the explosion have subsided? Would the cars run?

It was to find the answers to some of these questions that a group was invited to observe an experiment at the Nevada Proving Ground of the Atomic Energy Commission.

This group – which became the SAE-FCDA Advisory Committee – wrote the accompanying report, which tells about the experiment and some of the things that were learned.

A. L. Haynes of Ford is chairman of the committee. Other members are: R. B. Alexander of Packard, William Christensen of Nash-Kelvinator, V. B. Corey of Willys, N. F. Hadley of Chrysler, E. J. Martin and M. V. Muzzy of GMC, W. W. Smith of Studebaker, A. D. Wagner of Hudson, and R. C. Sackett of SAE, secretary.

The Author

A. L. HAYNES (M '42) is now assistant research engineer of the Engineering Research Department of the Ford Motor Co. He received his B.S. degree in Mechanical Engineering from Carnegie Institute of Technology and his M.S. degree in Engineering Mechanics from the University of Michigan. Prior to joining Ford, Mr. Haynes worked for General Motors Corp., Curtiss-Wright Corp., and Jones & Laughlin Steel Corp.

ions, and recommendations that will be of value to CDA organizations throughout the country.

On Tuesday, March 10, a week prior to St. Patrick's Day, when the blast was to take place, the members of the committee finally gathered in Las Vegas.

Fabulous Las Vegas, the gambling center of the world, is the nearest point to desolate, ominous looking Mercury, 70 miles out in the desert.

Mercury is the headquarters of the AEC, where all the details are worked out for the testing that goes on out at Yucca and Frenchmen Flats, 30 miles farther in past the entrance gates. (See Fig. 1.)

Three committee members had company cars that could be used for the daily 200-mile grind to Yucca Flat and return. The one-way 100-mile trip was knocked off in a little over an hour and a half – or somewhat over three hours per day were spent in traveling back and forth. (See Fig. 2.)

After getting final clearances and badges arranged the committee got down to the business of gathering the vehicles that came from many states and from two main sources. The major automotive manufacturers gave a group of cars and the National Automobile Dealers Association donated a number of cars. Three post office trucks were tested for the Post Office Department. The AEC provided a few "clunkers" that were placed as far up front as was deemed advisable. There were some 51 vehicles that went through the test.

The cars were gathered in the big Helldorado enclosure in Las Vegas. This is a fenced-in area for the purpose of holding rodeos and "Last

ATOMIC DEVICE TEST
YUCCA FLATS, NEVADA - MARCH 17, 1953
PRE-TEST VEHICLE INFORMATION

Report No. 1

1.- Test Vehicle Number 18. Odometer Reading 61939

2.- Make of Vehicle NASH. Serial # _____ Engine # K 117510

3.- Year Built 46. Model 4648 600. Body Type 4 dr. Color Dk. Blue

4.- License No. CAL 1U7953. Origin of Vehicle Southeast Nash Huntington Park, Cal.

5.- Condition of Vehicle:

(Indicate condition as far as possible by check mark)

Part Name	Excellent	Good	Fair	Poor	Remarks
Finish - Interior					
Finish - Exterior		x			
Structural		x			
Mechanical				x	No Brakes
Upholstery				x	Very Bad
Glass					Driver Door Glass Broken No regulator handles on both rear door glass
Tires		Front		Rear	
Radiators					Inlet at tank bad Core O.K.
Undercoat					
Engine					
Misc.					No Hubcaps

Running condition of Vehicle: Drivable

General Comments: Handmade Fabric Top

Date 3/12/53

SAE-FCDA ADVISORY COMMITTEE

Fig. 8 – Test sheet used in recording pretest vehicle data

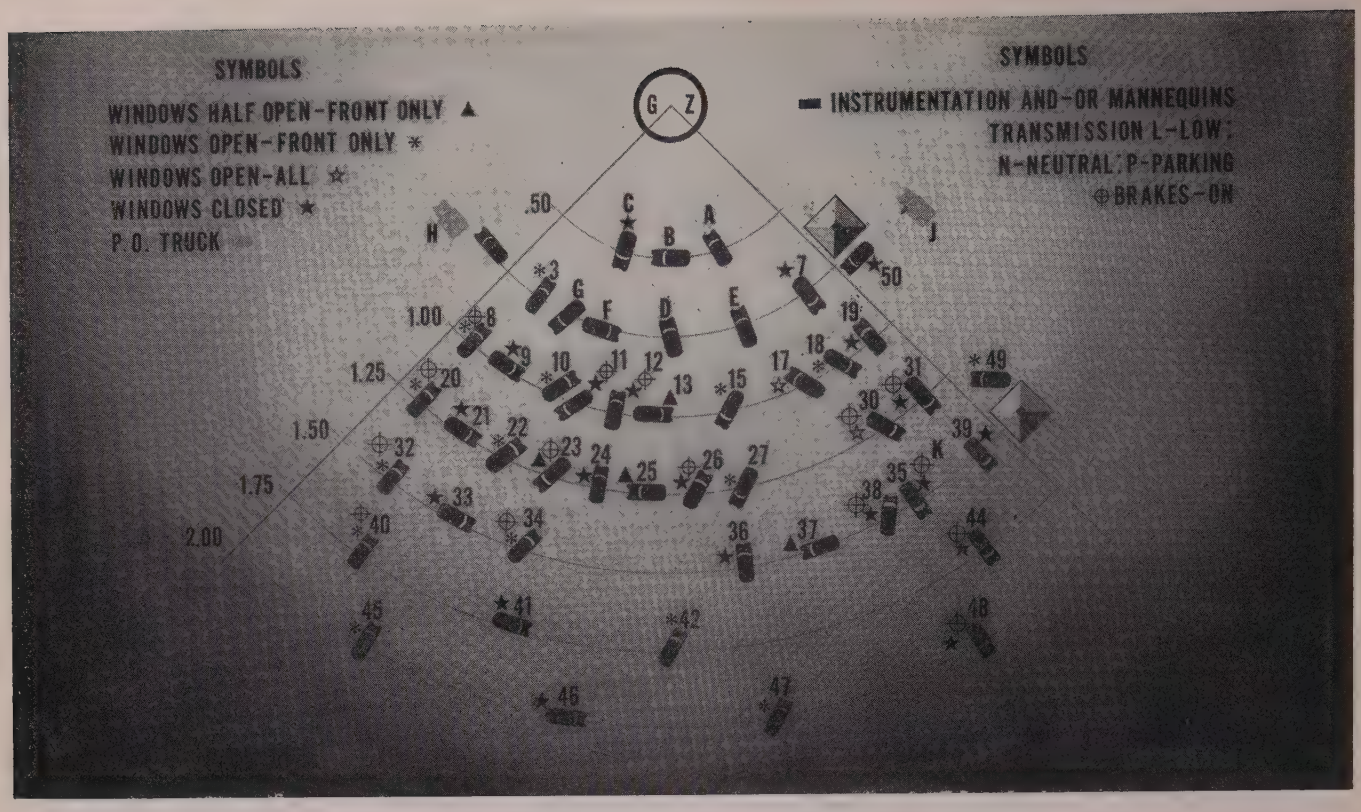

Fig. 9 – Schematic layout of automobiles for atomic bomb test at Yucca Flat, Nev.

Frontier" shows. The cars were numbered and inspected and a pretest data sheet was filled out for each automobile. (See Figs. 3-8.)

Finally, on Friday, March 13, some fifty students of a high school auto driving class drove the test cars out to the entrance of Mercury. They were chaperoned out there by a two-car Nevada State Police escort.

The committee, with the help of the AEC and FCDA people, drove the test cars the remaining 30 miles to the Yucca Flat site and placed them in position.

The SAE committee assisted in planning and evaluating the test for the March 17 shot. One member of the committee served in an advisory capacity in the other tests, which are still of a classified nature, hence cannot be discussed at this time. They will, however, be fully reported later.

In placing cars for the test many conditions of driving and parking were simulated. All cars were arranged as nearly as possible along arcs with ground zero as the center. Arcs were a quarter-mile apart and cars were generally about 50 ft apart along the arcs. First arc was 0.5 mile from ground zero and the last arc at the 2-mile radius.

Cars faced toward, away from, broadside to, and at angles toward and away from ground zero, as shown in Fig. 9.

Some had windows open, some partly open. Some had full tanks of gas, some half full, some almost empty. Some were in gear and some had the brakes set. These conditions and positioning were all prearranged by the committee.

One car was placed close to each of the two test houses – one at 3500 ft, the other at 7500 ft from the tower. The cars were behind the first house and at an angle in front of the second house. One post office truck was beside the house at 3500 ft from ground zero.

In 24 cars FCDA project personnel had placed fully clothed mannequins.

As shown in Fig. 10, AEC placed temperature gages and dosimeter and neutron badges, to measure radiation, in different locations on cars,

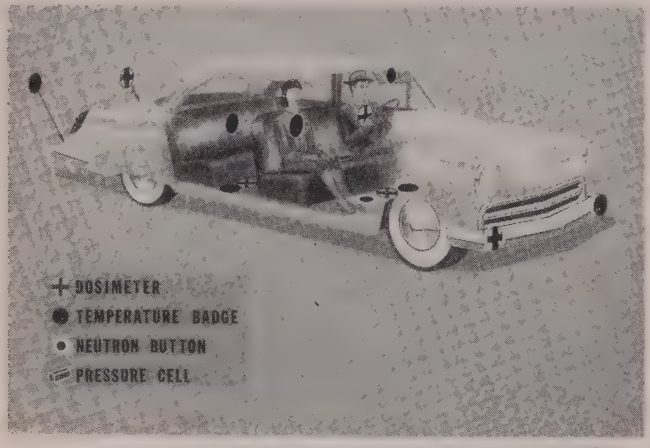

Fig. 10 – Locations in cars where instruments were placed

such as seat backs and cushions, floors, steering wheels, bumpers, and sun-visors. A few pressure gages of an experimental type also were used. These were to be collected as soon as it was possible to get into the area after the test and to be forwarded to Washington for processing. This work was finished by late Saturday afternoon.

On Monday, March 16, one day before A-Day, the committee left Las Vegas at 7:30 a. m. for final touches to the positioning of cars, the arranging for pretest photographs and all other last-minute details. (See Figs. 11 and 12.) The committee got back to Las Vegas at midafternoon for resting purposes.

At 10:00 p. m., with practically no sleep, the committee was off again for the test site. Although light clothing had been in vogue during the day, sweaters, jackets, and overcoats were the order of the night, because at 5:00 a. m. the chill of the barren desert was penetrating.

After a 1:30 a. m. breakfast at the mess hall in Mercury, the committee reached the vantage point at about 2:45 a. m. The last 2½ hr were full of excitement and anticipation.

Above the ledge of News Knob rises a rocky, precipitous slope on which newsreel cameras, news photographers, radio and TV people set up shop. At 5:05 a. m. came the deep ominous voice of the announcer with a 15-min warning. He told everyone not to look directly into the blast for the first

Fig. 11 – Some of test cars in position 1.5 miles from ground zero, before blast

instant. There was danger of seriously injuring the eyes at our distance of six miles from ground zero unless special goggles issued to us were worn.

At 5:10 a. m. the 10-min warning . . .

At 5:19 the 1-min warning . . .

The hush in the crowd gave you a creepy feeling and made the chilly air feel colder. You could have heard a pin drop on that ledge. And then the count – 10 sec. . . 9. . . 8. . . 7. . . 6. . . 5. . . 4. . . 3. . . 2. . . 1.

Looking 90 deg away from the direction of the tower at the zero second, and even through those

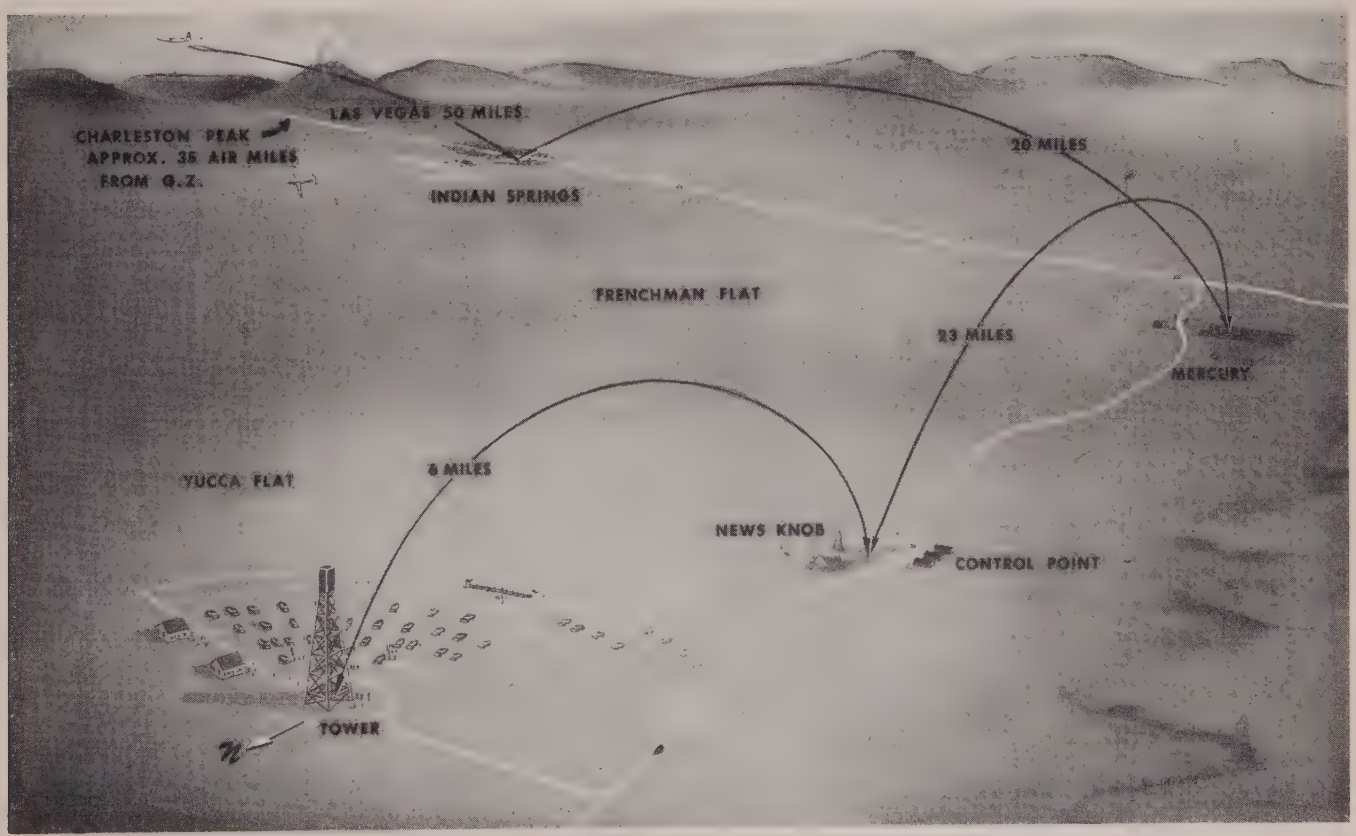

Fig. 12 – Map of area showing test cars in relation to test houses and foxholes

Fig. 13 – Committee on inspection tour after blast, dressed in rad-safe clothing (left to right): R. C. Sackett, V. B. Corey, W. W. Smith, E. J. Martin, A. L. Haynes, R. B. Alexander, N. F. Hadley, W. Christensen

ATOMIC DEVICE TEST — Report No. 2
YUCCA FLATS, NEVADA - MARCH 17, 1953
AFTER TEST DATA

1.- Test Vehicle Number 18 . Make of Vehicle 1946 Nash

2.- Station (Miles from Ground Zero) 1 (Zero)

Orientation

	Left Ft.	Right Ft.	Left R.	Right R.	Windshield	Rear Window
3.-Windows - Position	O	O	C	C		
Damage	Shattered		Shattered			
4.-Wings - Position	O	O	C	C		
Damage	Shattered		Shattered			
5.-Tires - Inflation						

6.- Radiator:- Level of Coolant: Visible _____ : Invisible _____

7.- Battery (Condition):- Water - Visible _____ : Invisible _____

8.- Is Vehicle Drivable? - Yes ____ No. X ___ If "No", why? Dead battery

9.- Distance Vehicle Was Moved and Attitude Rear moved about 1 ft. away from original position.

10.- Finishes:

(a)-Interior _____

(b)-Exterior L. side scorched.

11.- Upholstery, Trim and Insulation Trim loose

12.- Plastics:

(a)-Interior _____

(b)-Exterior _____

13.- Sheet Metal Appearance:

(a)-Top Caved in

(b)-Door Panels L. side caved in

(c)-Fenders R. R. blown out.

(d)-Hood _____

(e)-Rear Deck Caved in a little.

(f)-Gas Tank _____

Fig. 14 – Test sheet used in recording after-test data

thick black goggles that made a bright headlight look like a tiny firefly, the landscape for miles around suddenly became as bright and as brilliant as on the sunniest day in June, perhaps even brighter. It was beautiful, horrible, terrifying – all in one. Dante would have had a difficult time describing that inferno.

At 1/10 millisec after time zero the luminosity as observed at a distance of 5.7 miles was 100 times that of the sun as seen from the earth's surface.

The temperatures at the center of the explosion reach the levels of those at the interior of the sun – over 1,000,000 deg and, of course, the tower was completely vaporized. The ionization right after the blast was a display never to be seen on the Fourth of July. And then the huge pillar of radioactive dust mushroomed up across the flats.

On Wednesday morning, the day after A-Day, Dr. Corey and the author put on man-from-Mars rad-safe clothing – used especially for radiation protection – and teamed up with FCDA and AEC personnel, together with a monitor and a photographer (Fig. 13). We were assigned an army weapons carrier for the purpose and proceeded well toward the center of the area. Our stay in the area was limited to 2 hr. Anything over this would have been in excess of the exposure limit of 1½ roentgens we had set for ourselves that day. Since we wanted to go in again the following day, Thursday, and because we were held to a total exposure of 3 roentgens as a safe limit, we left the area after 2 hr on Wednesday and after about 3 hr on the second day. On the second day the full committee, dressed in rad-safe clothing, went into the hot area with a monitor and a photographer and surveyed the cars. Incidentally, 400 roentgens are considered a median lethal dose of radiation – 50% of those exposed to such a dose would be expected to die. Post-test data reports (Fig. 14) were also completed at this time.

Physical Damage to Cars

The following observations are given strictly as opinions of the SAE committee. The committee is still working with FCDA on official interpretations. It should be emphasized here that all the observations and opinions are made on the basis of one test – the March 17 burst. It was a tower shot and it was held out in the open desert.

1. There was no significant structural damage to frames, front and rear suspensions, motor mounts, or other structural parts. There was no apparent structural damage even when cars were rolled over. (See Figs. 15-18.)

In general, cars that were oriented broadside to the blast overturned at ranges comparable to less than 0.75 mile from the 15-kiloton tower shot of March 17.

Cars that could be operated before the burst

would operate after the burst, provided they did not burn.

(The March 17 shot has been referred to as a "baby bomb," or a "low power" device. Such phrases are entirely misleading. For example, a frame house located 3500 ft from ground zero that was almost completely demolished by the 15-kiloton shot would have sustained the same damage a very short distance farther away from a 20-kiloton "nominal" bomb.)

2. Glass breakage was common at the closer ranges. In terms of the March 17 shot, most glass breakage occurred within 1.25 miles from the tower.

Safety plate glass at near ranges was blown out in large pieces, which were badly cracked and crazed but which did not shatter. However, not all windows in any car broke, in most cases. There were many instances where safety glass gave under the blast impact but did not entirely blow out of its frame. In a few cases, only a relatively small hole was opened in the center of the caved-in glass.

Tempered glass of the type commonly used in rear windows of passenger cars did not splinter but broke into countless small particles, some of which were blown a considerable distance at apparently high velocity. Curved windshields and rear windows generally resisted failure from blast better than small or similar size flat glass surfaces. (See Figs. 19-21.)

3. Based on the March 17 test, the percentage of cars lost to fire was surprisingly low. Where fire occurs it apparently begins with smoldering of seat covers, head liners, or door panels.

There was some scorching of paint, tire sidewalls, and other exposed surfaces. And, as expected, the darker colors generally were more susceptible to heat effect than the lighter ones.

There was evidence that some paint softened from heat flash, allowing sand and gravel from the blast wave to impregnate it.

There was no evidence whatever that the gasoline tanks or fuel systems contributed to the initial fire hazard. (See Figs. 22 and 23.)

No gas tanks exploded, even where the car was ultimately burned out entirely. This was true regardless of the amount of gas in the tank; amounts varied from ⅛ tank to full. Nor did it seem to matter whether or not the gas tank was capped or open.

4. There were no apparent physical effects on cars from the initial radiation of the bursts. Some cars were exposed to "fall-out" (dust being rained out or settled out from the cloud, which is formed after the shot) and consequently were contaminated.

All cars, when removed from the firing area, were subjected to radiological safety procedures. These included careful monitoring of the entire car to determine the levels of radioactivity, then decontamination, if necessary. In some cases decon-

Fig. 15 – Committee members inspecting damage to car No. 36, located 1.5 miles from ground zero, facing ground zero, with all windows closed. No glass broken, roof caved in badly, both right doors dented, left front door caved in

Fig. 16 – Car No. 38, located 1.5 miles from ground zero, with all windows closed. No glass broken, roof caved in badly, door panels caved in, molding ripped loose

Fig. 17 – Car No. 22, located 1.25 miles from ground zero, at an angle toward tower, with front windows open and rear windows closed. No glass or other apparent damage, except left front door was loosened from hinges

Fig. 18 – Car No. 24, located 1.25 miles from ground zero, facing ground zero, with all windows closed. Glass shattered but in place, top caved in badly, door panels caved in

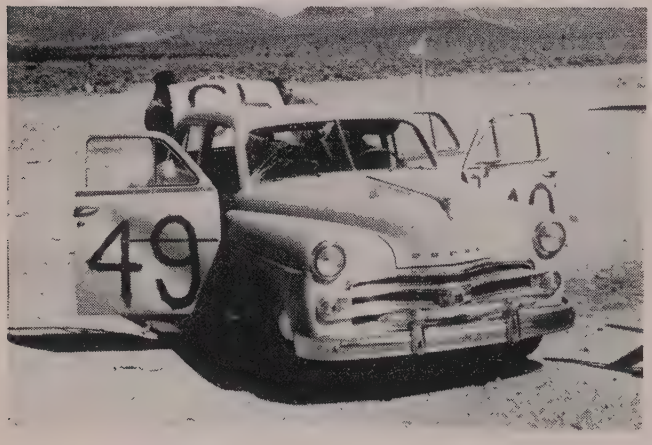

Fig. 19 – Car No. 49 in front of second house, located 7500 ft from ground zero. Right side of windshield blown in, large piece of unsplintered glass on front floorboard, finish cracked, paint fracture, trim blown off, top caved in 10-12 in., fracturing heads of both front-seat occupants. Both right doors caved in, left rear door open, trunk blown open, latch sheared off

Fig. 20 – Committee members inspecting damage to car No. 7, located 0.75 mile from ground zero, facing tower, with all windows closed. Windshield broken and gone, rear window completely gone, glass scattered 20 ft in front of car, right window blown in. Car started and was drivable. Seat backs scorched, boy's jacket burned off in rear seat, woman's clothing scorched and blown off, inside trim blown loose, rubber seals scorched, top caved in 12 in., left door panel dented, right door panel caved in, right fender rear panel dented, trunk lid caved in, gas tank loose and one strap broken

Fig. 21 – Car No. 46, located 2 miles from, and broadside to, ground zero, with all windows closed. No glass damage, no indication of heat damage. Two transverse depressions 4-6 in. in top on either side of center pillar and top rib, left front door panel dinged slightly

Fig. 22 – Car No. 10, located 1 mile from, and at an angle facing, ground zero. Interior condition of car, upholstery, and the like, was bad before blast, fiber seat covers, windows were cracked when car was received, recap tires. Car interior burned out, glass melted, tires burned out, car standing on rims, paint burned

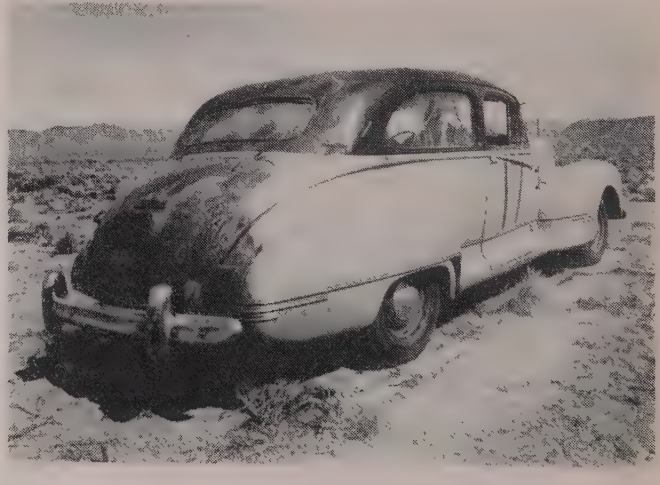

Fig. 23 – Car No. 17, located 1 mile from ground zero, oriented diagonally toward ground zero, with front windows open and rear windows closed. Left rear window cracked, right rear window shattered, paint scorched, except where protected by license plate, trim strips loosened, top caved in, right side dented in some, hood loosened

tamination was carried out simply by washing the car. In others, upholstery and interiors were vacuumed.

Some cars required minor decontamination because project personnel had carried radioactive material into the cars on their feet. All cars were successfully decontaminated and declared to be "rad-safe" – that is, a radiological safety officer declared that they did not constitute a radiological safety hazard of any kind.

5. There was no significant difference between makes or models of cars, with the exception of wooden-bodied station wagons. As might be expected, these were less blast-resistant. (See Fig. 24.) No convertible models were tested.

The only tire failures were due to fire, with the exception of one blowout caused when a post office truck overturned.

In cases where car windows were closed, the most general damage consisted of dished roofs and side panels. In the March 17 shot, cars with closed windows at the 1.75- and 2-mile zones suffered dished tops; cars at these distances with windows open suffered no important damage. (See Figs. 25 and 26.)

The big, square-bodied post office trucks did not react differently from the passenger cars, with the single exception of a truck oriented broadside to the blast. This was blown over at the 0.75-mile range during the March 17 shot, while all passenger cars, except a station wagon in the same cluster, remained upright.

Hoods and trunk lids were sometimes blown or sprung open. Some doors were jammed (as shown in Fig. 27), making a quick exit difficult.

Implications of Physical Damage to Cars

(1) Cars are useful as vehicles after the blast; (2) there are no significant differences between makes or models of cars; (3) what remains is the question: What would happen to humans if they were in the cars?

The effects on vehicle occupants must be evaluated in terms of the three damaging components of an atomic burst: (1) heat-flash, or thermal radiation; (2) nuclear radiation; and (3) blast. These evaluations are still in process and will be issued formally in the final report; however, it might be well to discuss some of the considerations involved.

Since the angle at which heat-flash enters a car changes according to the height of the atomic burst, it stands to reason, the closer to the floor an occupant of a car is the more protection he has. In air bursts an occupant might be burned on limited exposed parts of his body while receiving general protection. Clothes would, of course, give additional protection.

As was mentioned earlier, mannequins, all of which were seated upright, were installed in a

Fig. 24 – There was no significant difference between makes or models of cars, with the exception of wooden-bodied station wagons. As might be expected, these were less blast resistant. There were no convertible models tested.
The only tire failures were due to fire, with the exception of one blowout caused when a post office truck overturned.

Fig. 25 – Car No. 48, located 2 miles and facing away from ground zero, with all windows closed (left windshield had two stars in it when received). Only glass damage was shattering of left windshield, top caved in, door panels pushed out, exterior trim strip on right torn loose

Fig. 26 – Car No. 41, located 1.75 miles from, and broadside to, ground zero, with all windows closed. No glass damage, although both sides of windshield had rock stars, top caved in 8-10 in. in center, left front door caved in slightly, left front outside molding pushed off, dome light lens popped off

Fig. 27 – Car No. 25, located 1.25 miles from, and broadside to, ground zero, with front windows one-half open. Rear window blown out, side caved in

number of cars for the March 17 test. Some showed heat-flash damage within 0.75 mile from the tower. The damage consisted of charred or scorched clothing on the upper parts of the body. In most cases the damage was confined to the shoulders and above. The "skin" surfaces of the mannequins were not of a nature to show thermal effect easily, but human skin at the same distances might have been severely burned. (See Figs. 28 and 29.)

The fire that started because of the effect of thermal radiation on the upholstery progressed slowly. There was little difference between this fire from heat-flash and those that are sometimes caused by dropping lighted cigarettes onto upholstery.

Another factor to be considered here is nuclear radiation.

Most residual radiation comes from contaminated particles of matter. Closed windows and doors on a car would help to keep these particles out. Further, because of the nature of residual radiation, it is somewhat less penetrating than initial radiation and the steel body and floor of a car do offer some protection from initial radiation if they are within the area of high concentration. However, radiation dosages are also a function of time.

We have seen the effects of the blast on the cars themselves. We will now cover the question of the effects of the blast upon the occupants.

On the basis of the March 17 test, it might be said that passenger cars would provide better protection from missiles or debris than the above-ground floors of frame dwellings, except where such dwellings have properly prepared shelter areas.

No comparison should be drawn between the protection afforded by cars and the basements of houses; there are too many variables. For example, persons in cars at the 0.5- or 0.75-mile range during the March 17 test might either have been harmed by blast or possibly by a lethal dose of nuclear radiation. On the other hand, persons in the basement of a test dwelling 3500 ft from ground zero would, even had there been no shelters, have been protected from nuclear radiation. Beyond the range of casualty-producing radiation, persons in cars would have been generally safe, had they been on the floors or lying on the seats. But they would also have been safe in a basement – provided the basement had access to the outdoors in case of fire.

As a general rule, any below-ground refuge should be preferred to a vehicle.

The "dishing effect" on the steel tops of cars with windows closed was one of the most dramatic aspects of the test.

In the March 17 test all test cars with windows closed suffered dished tops, regardless of distance from ground zero. At the 0.5- and 0.75-mile ranges, some cars had this type of damage whether windows were open or closed. Beyond 1 mile dishing

Fig. 28 – Interior of car No. 3, located 0.75 mile from ground zero and facing ground zero, with front windows open, rear windows closed. Windshield shattered but still in place, radiator water visible, bad rubber flashes around windows, seat covers burned, top caved in

Fig. 29 – Interior of car No. 49, located in front of second house, 7500 ft from ground zero, showing head of mannequin against top

Fig. 30 – Car No. 50, located in rear of house No. 1, 3500 ft from ground zero, with all windows closed. Car in bad shape, house roof blew over onto it

Fig. 31 – Committee member inspecting damage to car No. 50, shown in Fig. 30

was generally restricted to cars with windows closed.

There is some evidence that closing the window may help to prevent glass damage beyond the ranges where some glass damage is inevitable. This is because the window has one unsupported edge. However, such aid against breakage would apply only in a limited zone. Closer to the burst, all glass would have a chance of failure. Farther from the burst, no glass would fail.

There is also evidence that the dishing of the steel top cushions the shock effect somewhat in the interior of the car. But again, the protection value is doubtful since at ranges where blast pressure exists capable of direct harm to a human being, glass would be damaged and some tops dished in whether or not windows were open.

In the first test soldiers were in trenches 2 miles from ground zero; one cluster of test cars was at the same distance. The soldiers suffered no harm from the blast, while cars with windows closed had the tops caved in.

There were several instances when cars were overturned by the blast. Apparently, it made no difference whether windows were open or closed.

It may be concluded, therefore, that the position of car windows makes no great difference in terms of personal protection. However, the March 17 test indicates that occupants of cars should get as low as possible to avoid not only heat-flash and flying glass, but the dishing of tops. Mannequins seated upright in test cars suffered broken heads when the tops dished.

The cases where vehicles were parked beside or behind buildings (Figs. 30 and 31) indicate that cars near buildings might be endangered by debris from collapsing structures. In addition to this observation, there is the rather frequent experience in cities of cars being smashed by falling masonry or trees during high winds.

What conclusions can be drawn from the above observations?

1. In damage sustained there were no significant differences between makes or models of vehicles.

2. Cars appear to be generally fire-resistant, but some fires may start in fabrics. However, such fires start slowly and could be extinguished by occupants in the same way that ordinary fires are put out.

3. Cars provide good shelter if they are beyond the major damage radius of an atomic burst. At comparable distances, they appear to offer better protection from flying debris than the above-ground floors of frame houses in which no prepared shelter space is available. This is due to the fact that an automobile does not produce as many missiles which might cause casualties and gives some protection against missiles and debris from outside sources. Under any circumstances, a properly located and prepared shelter, or shelter area such as those already marked by Civil Defense in many cities, is to be preferred to a vehicle.

Recommendations to the Motoring Public

In the Spring series of tests there were over two hundred car exposures. These will be subjected to a detailed statistical analysis by the SAE, and biomedical experts of FCDA and AEC before final conclusions or recommendations are made to the motoring public.

Acknowledgment

Finally, I would like to acknowledge and commend the Federal Civil Defense Administration on the splendid cooperation they extended to us in carrying out this experiment and on the important work it does in keeping the "whole citizenry alerted to their collective tasks."

Uncontrolled Combustion

THE major purpose of this note is to discuss the fact that preignition is neither a product of the atomic age nor only of the high-compression-ratio engine. Preignition is an old, if not well-known, disease of the Otto-cycle engine despite the thought in some quarters that it is a recently developed engine ill.

The note is the result of a request by C. J. Livingstone that the author attempt to clarify the terms used to describe the various forms of uncontrolled combustion that occur in spark-ignition engines. Livingstone, in particular, was disturbed by the fact that the term "preignition" is now used to describe several types of disordered combustion. The author was requested to write the note on the grounds that he was working on runaway, destructive preignition in aircraft engines in World War I. The title of the note is probably badly chosen.

Uncontrolled combustion is used in the sense of ignition of the charge which is not the direct result of the spark. Thus, knock is uncontrolled combustion since it is an indirect result of the spark. The definitions used below are the author's, although some of them have been widely used. Some of the definitions may be objected to but are used since they appear to be descriptive at least.

1. *Knock or Detonation* – Knock may be defined as spontaneous ignition of a portion of the end gas as a result of its temperature and pressure. This ignition occurs in the end gas prior to the arrival of the flame which has been initiated in the rest of the charge by the spark. This ignition of the end gas can be and often is described as "auto-ignition." Knock can occur in an engine in which no spark is occurring and in which part of the charge is ignited either by its pressure and temperature or by some hot surface. Knock occurring in conjunction with ignition from a hot spot can be a stable condition of operation and one that does not result in shutting the engine down due to loss of power or to backfiring. Knock in the absence of spark is not important and is usually only experienced on single cylinders in the laboratory. The spontaneous ignition of the end gas is commonly called detonation, which it really is not. The ignition is very rapid and its suddenness may be compared to the explosion of a rifle cartridge, which burns with great rapidity but does not detonate. If the rifle cartridge detonated, the rifle barrel would be shattered.

2. *Preignition* – Preignition is a term that is currently applied to cover several varieties of uncontrolled combustion. The term is here used in the sense in which it has been understood by the aircraft-engine engineer for many years and meaning ignition of the charge occurring prior to the passage of the spark. Preignition as it is known in aircraft engines is almost always the result of ignition from a hot surface within the combustion chamber. It is almost always an unstable condition in that, once it starts, the surface causing ignition rises in temperature with each cycle and, as a result, ignition occurs progressively farther and farther down the compression stroke. Eventually, if the process proceeds, ignition will occur while the intake valve is still open and the engine will backfire. Preignition in aircraft engines is usually very destructive and dangerous. As a rule it causes seizing or melting of a piston. As a result of backfiring it can cause explosions and fires in the induction system. The effects of this type of preignition are dealt with in more detail later.

It should be emphasized that preignition is mostly not audible and need not be preceded by knock. In many cases an engine will go out of normal combustion straight into violent and destructive preignition without the occurrence of audible phenomena. Knock, by increasing the temperature of the surfaces within the combustion chamber, can cause preignition.[1]

3. *Autoignition* – Autoignition may be defined as any form of ignition that is not the direct result of the spark. Thus, the spontaneous ignition of the end gas under knocking conditions (1) above, or preignition (2) above, are forms of autoignition. It may or may not be the result of ignition from hot surfaces within the combustion chamber. It can occur with dead-clean and well-cooled combus-

[1] One inventor, working with a CFR engine without power measurement, developed a method of suppressing knock by means of "activated" surfaces. Using normal heptane, he increased the compression ratio until knock became very violent. Further increase of compression ratio caused the knock suddenly to disappear. When the experiment was repeated with the CFR engine coupled to a dynamometer, the disappearance of knock was found to coincide with negative indicated power, and this power level did not change when the ignition switch was cut. The surfaces were "activated," they were hot.

in Spark-Ignition Engines

S. D. Heron, Ethyl Corp

This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1953.

tion chamber surfaces if the temperature and pressure of the charge are sufficient for ignition of the particular fuel used. This ignition, with clean, well-cooled combustion-chamber surfaces, may or may not be accompanied by knock (a compression-ignition engine, of course, functions by autoignition of this type) depending upon the type of fuel used.

The term autoignition is here used in a much more restricted sense than the above and covers three types of uncontrolled combustion, as follows:

a. Wild ping: This is a condition caused by combustion-chamber deposits which induce violent knock for a few cycles. The cycles of knock may be followed by a few cycles of preignition (without audible knock), which in turn are followed by nonpreigniting combustion with or without knock. Wild ping does not necessarily develop into cyclic preignition. Wild ping is frequently referred to as preignition, but it appears that it can result from ignition which occurs subsequent to the passage of the spark.

Wild ping has so far not been of known concern to the aircraft-engine engineer, although it probably occurs. It may be followed by a few cycles of preignition, which will probably result in purging the deposit,[2] causing the preignition, but this is not generally a cause for concern unless it shows up in increased cylinder temperature.

b. Afterfiring: This is a term used for many years in the aircraft-engine field to describe firing that occurs continuously or intermittently at the prevailing load and speed when the ignition switch is cut for a few cycles. It is often a sign that an aircraft engine is not far from the onset of preignition and that increase of throttle opening for reduction of fuel/air ratio will lead to preignition. Afterfiring has no relation to continued firing, which may occur at closed throttle and low speed after cutting the ignition, and which is discussed later.

Afterfiring can be continuous and the engine may pull full power for several minutes after cutting the switch. This may, in some cases, be a stable condition which has no tendency to develop into runaway preignition.

Afterfiring, of an intermittent variety, often occurs with spark plugs having platinum fine wire electrodes when the plug is far below its preignition limit.

c. Afterrunning: Afterrunning may be defined as continued firing at the idle condition after the ignition is turned off. Afterrunning seems to be currently mostly a concern of the passenger-car engineer. The British refer to it as "running on." In the main, afterrunning is a product of combustion-chamber deposits. Deposits are not essential to its occurrence, however, since it can and does occur with dead-clean combustion chambers.[3]

Afterrunning is of little importance in the aircraft field since, in many cases, it is prevented by the "idle cutoff" in the carburetors, which cuts off the fuel supply to the idle jets.

Afterrunning is at times referred to as preignition, but this would seem to be distinctly a misnomer since it does not precede any other type of ignition.

This history of preignition is largely based upon the author's experience with it in the aircraft-engine field and beginning in 1917. Preignition was known before 1917, as was the difference between preignition and knock. Prior to 1914 preignition had been a matter of much interest to builders of European racing cars and racing motorcycles. This interest was evidenced by their efforts to

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[2] Preignition in aircraft engines, if caught in time, will usually result in purging the deposits from the aluminum surfaces of the combustion chamber. These surfaces often are dead clean after a short burst of preignition. Preignition of the runaway variety has been deliberately used as a means of completely purging deposits (except from the exhaust valve and spark plug) in cast-iron laboratory engines. Such preignition for a period of about 30 sec does an excellent job of purging cylinder-wall and piston-crown deposits during fuel studies and spark-plug fouling tests. Successful use of this technique has involved an imep of about 350 psi at 2700 rpm, but most engineers are afraid to try it.

[3] Reliable reports from Europe indicate that overlapped valve timing can promote afterrunning. At first thought this appears puzzling but, on consideration, it seems that overlap will increase the compression pressure and temperature at idle.

secure suitable spark plugs, although it is not clear that the builders knew that their troubles were due to preignition. They may have merely known that the engines would not run at high power without suitable cold spark plugs. Many of the then current spark plugs would melt if run at high power in racing engines.

In 1917 the Rolls-Royce watercooled aircraft engines began to suffer from preignition, which would burn away one-third of the head of an exhaust valve in about 5 min running at full power. The engines were of about 4½-in. bore, ran at about 1800 rpm, had a compression ratio of about 5½/1, and developed a brake mean effective pressure of about 130 psi, using fuel of about 55 octane number. The trouble was partly due to the then current mica spark plugs, and Rolls demonstrated this by building an engine with special cylinders whereby the spark-plug nose temperature could be observed through the open exhaust valve (no exhaust stacks). The author saw this demonstrated late in 1918 and observed that when the mica became white hot on the surface, the exhaust valve temperature rapidly rose to 1700-1800 F. This test did not go to the end point of burning a large segment out of the exhaust valve since the test was terminated by a mechanical failure elsewhere in the engine.

Rolls-Royce put out a pamphlet discussing this preignition difficulty and stating that engine failure due to a burnt exhaust valve was the lesser evil since, if the exhaust valve did not fail first, the piston certainly would (see Appendix).

For several years after the Rolls-Royce difficulty, not much attention was paid to preignition in the aircraft-engine field. This at least was true in the United States. It would now seem certain that some of the unexplained cylinder and piston failures must have been due to preignition resulting from spark plugs of very uncertain heat rating and fuel that varied all over the place down to as low as 40 octane number.

With the introduction of leaded aviation fuel in 1928, the U. S. Army Air Force perforce began to pay attention to preignition. Lead deposits in the combustion chamber were responsible, but in a secondary manner, since preignition resulted from reaction products of the lead deposits with the materials of construction used in the combustion chambers. Preignition difficulties resulted in part from these reaction products and in part from either the considerable increase of engine output which lead permitted or from the considerable increase of cylinder temperature at constant output which was permitted by increased octane number. Examples of increased cylinder temperature at constant output were due to an increase of jacket temperature from 180 F (water) to 300 F (ethylene glycol) or reduced cooling air supply in aircooled engines (tighter cowling).

The two major causes were exhaust valves and spark plugs. The mica spark plugs were of uncertain heat rating (there was no adequate method of determining heat rating and heat rating varied widely in a given plug model) and lead compounds reacted vigorously with both the mica and with electrode material. Leaded fuels, more than any other single cause, eventually killed the mica plug.

The primary cause of the exhaust-valve difficulty was the formation of a coating of black magnetic iron oxide (Fe_3O_4) on the surface of the head of the exhaust valve, which was exposed in the combustion chamber. This deposit was granular but nevertheless tightly adherent and it contained practically no lead compounds. The formation of the oxide layer was accelerated in some but not all cases by attack of lead deposits upon the seating face of the valve. The attack upon the seat led to valve leakage, which increased valve temperature. In the case of one aircooled engine having solid exhaust valves, preignition would start within 30 hr of beginning a test with new valves. If the preignition was detected prior to engine damage, the deposit could be removed by sandblasting and the engine would then run for about 20 hr before preignition again set in. One engine was repeatedly treated this way without engine failure. One engine of similar type failed three times between 40 and 45 hr on an attempted 50-hr type test. The failures were all due to melted or seized pistons. In one case a cylinder barrel went from normal temperature to visually red hot in about 30 sec. This engine was made satisfactory for this power rating by adopting sodium-cooling of the exhaust valve without change of valve material.

Another engine of the liquid-cooled type and having sodium-cooled valves was in severe difficulty with burned valves. The burning difficulty was cured by adopting stellite facing of the exhaust valves. This facing then became general practice for all exhaust valves in high-powered aircraft engines. The stellite facing then enabled the engines to run long enough on type test so that preignition resulted from iron oxide scale on the exhaust-valve crowns. This was in part cured by the adoption of exhaust-valve material with greater corrosion resistance.

From 1932 until the present the aircraft-engine exhaust valve has ceased to be a significant source of *preignition* difficulty (which is not saying that the valve itself is not still a source of difficulty). This elimination of preignition difficulty has been the result of improved cooling and improved materials, hollow-head sodium-cooled valves with the top of the crown coated with 80 nickel-20 chromium (Brightray – introduced by the British in the middle 1930's) being an example. While the aircraft exhaust valve has ceased to be a source of preignition difficulty due to scaling, this trouble still arises at times in marine and truck engines. In cases where preignition arises from spark plugs or combustion-chamber deposits (see below), exhaust-valve stretching is a common result. In fact, the author has seen a sodium-cooled valve

PREIGNITION, Mr. Heron points out, is an old, if not well-known, disease of Otto-cycle engines.

Despite its long history, however, there is much confusion in the terms used to describe it and the other forms of uncontrolled combustion in spark-ignition engines.

The author has tried to bring some order out of the confusion by defining the three terms: knock, preignition, and autoignition.

He also relates his experiences with destructive, runaway preignition in aircraft engines during World War I.

THIS paper and the four that follow, by Melby-Diggs-Sturgis, Hirschler-McCullough-Hall, Williams-Landis, and Winch, constitute the complete Symposium on Preignition that was presented at the 1953 SAE Summer Meeting. Discussion of all papers is published at the end of the last paper in the group, by Winch.

The Author

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from a large marine engine which was both stretched in the stem and upset in the neck due to preignition.

The spark plug as a source of preignition difficulty is still with us despite very important advances in plug construction and in uniformity of heat rating of given plug models. The difficulty is, in the main, due to the steady increase in severity of operating conditions that has occurred since 1932. Fuel of 100 octane number put the quietus on the mica plug, 87 octane number had almost done so but the 30% (approximately) increase of output resulting from increasing octane number from 87 to 100 was the last straw. The mica plug would have gone anyhow because it could not have been produced in the quantities required in World War II. While the ceramic plug represented a great advance over the mica type, it did not eliminate plug difficulties due to preignition. Cracked and broken ceramic cores were and still are a cause of preignition. In the early ceramic plugs, cracks and breakage, due to thermal shock, and excessive stresses in the core, due to design, were frequent causes of preignition but have been largely eliminated. The ceramic plug is, unfortunately, still relatively fragile, and cracked and broken cores can be produced by dropping on a concrete floor or by a hard blow with a wrench. Fragility is being steadily reduced by improved design but the ceramic plug will still not stand the mechanical abuse that the mica type would tolerate.

The ceramic plug can still be a cause of preignition even when in perfect mechanical and thermal condition and when free of deposits. The heat rating of a spark plug used in an aircraft engine is always a nice balance between fouling at light load (particularly fouling during prolonged idle while awaiting take-off) and preignition at either cruise (very lean) or take-off.

The last significant cause of preignition in aircraft engines is the use of additive oils. Additive oils have never been widely used in aircraft engines of high power but they have had extensive use in the small engines of personal aircraft. Additive oils have never been permitted for military engines, except for the use of tricresyl phosphate as a run-in oil in an engine manufacturing plant. Additive oils containing corrosion preventives are used by the military services for engine preservation and engines are flown with these oils while the aircraft is being handed over by the contractor. These oils are often known as "flyaway oils." Tricresyl phosphate has never been shown to cause preignition either in run-in oils (preignition was experienced during run-in in one particular plant and at less than 50% of take-off power but was due to cracked or broken ceramic spark plugs) or in rather considerable commercial service on airlines. The corrosion preventive oils used for flyaway at one time were at least under suspicion in regard to preignition since certain types caused very heavy combustion-chamber deposits and much plug fouling difficulty.

It is generally known that the petroleum industry has spent a great deal of effort on additive (detergent) type oils for aircraft use and some companies have courageously published details of their troubles. Other companies have been almost ready to market after long experimental programs but refrained at the last moment when preignition reared its ugly head.

The additive types which are known to cause trouble are those which leave deposits of high melting point in the combustion chamber. Some types of calcium and barium compounds have been notable offenders. The deposits which cause preignition difficulty are mixtures or reaction products of the ashes left by the lubricating oil and by the leaded fuel. Whether the trouble would arise in the absence of lead in the fuel is not known since aircraft engines of high power are not operated on unleaded fuels. With leaded fuels and non-additive oils, the lead deposits do not appear to cause preignition except where the lead deposit reacts with materials of combustion-chamber con-

struction (that is, violent attack on unsuitable spark-plug ceramics, and so on).

Difficulties due to preignition with additive oils have arisen with high-powered marine engines. In one recent case of preignition in such an engine of the naturally aspirated type, additive oil is used and the cause of preignition is as yet obscure, although spark plugs definitely do not seem to be at fault. In this case, preignition causes severe attack on the piston crown but without seizure or melting. Preignition, in this case, is made apparent by backfiring and only the piston has to be replaced although the exhaust valve may be the cause of the difficulty.

One case of preignition which never became destructive (but which could have if operation had continued) occurred in a light-plane engine and with an additive oil which apparently produced a low melting point ash. The fuel in this case only contained 0.1 ml tel. Fused glassy deposit was found on the spark plugs and appeared to be low melting (but was not proved to be). Atmospheric dust (silica or silicates) may have been a cause in this case.

Destructive preignition of no commercial significance is currently experienced with engines used for a variety of racing purposes. Alcohol fuels are often the cause for two reasons. Firstly, alcohols are prone to preignition, secondly, they give no preliminary warning by knock or other audible phenomena and most drivers expect to be warned of fuel difficulties by audible effects. One European engineer with rather extensive road racing experience states that, except with the most experienced racing drivers, medium-cold spark plugs are desirable since they give warning by rough running and backfiring. With dead cold plugs rough running warns of the onset of preignition but the less experienced driver is only informed of its presence by a hole in the piston.

Experience with aircraft engines indicates a few variables which influence the tendency to preignition, and these are as follows:

a. The more sensitive the fuel the greater the tendency to preignite.

b. Increase of fuel/air ratio has a marked effect in suppressing preignition.

c. Retarding the spark has been shown to be very effective in reducing the tendency to preignition arising from spark plugs. This was determined in a water-cooled engine and has not, to the author's knowledge, been confirmed in aircraft engines. (It has been stated on reliable authority that spark retard aggravates "afterrunning." Spark retard, in general, reduces cylinder-wall temperatures and this may promote deposit accumulation, which will, in turn, promote afterrunning.)

Summary of Present Known Causes of and Effects of Runaway Preignition – Preignition as dealt with throughout this note is that occurring at reasonably high engine speeds, that is, 1500 rpm and upwards. This is, in the main, due to the fact that aircraft engines do not develop significant power at low rpm. Beyond the knowledge that preignition can cause backfiring at low rpm, the author is almost without knowledge of the effects at such engine speeds. *Destructive* preignition is, in general, a disease occurring at high rpm.

Causes:

a. Spark plugs. Plugs which are too hot for the engine condition of operation are a major cause. Cracked or broken ceramic insulators in plugs which are suitable when undamaged have been a potent source of trouble. A broken piece of core held from falling into the cylinder by means of the ground electrodes can have particularly vicious effects. In a similar case, where the ground electrode construction allows the broken piece to fall into the cylinder, preignition may not occur. In fact, such broken plugs in some cases have higher preignition ratings than an unbroken plug of the same type.

b. Additive oils. Additive oils of the detergent type can, if the additive is unsuitable, produce very serious difficulties in aircraft engines. Very dirty pistons will often go for long service periods without ring sticking. Pistons with silvery white skirts and ring belts may have a short and gay life, ending in a molten condition.

c. Exhaust valves. Exhaust valves, at least in high-powered aircraft engines, do not appear to be a present cause of preignition. With current practice, valve-crown corrosion is still encountered but the corrosion products do not appear to cause preignition.

Sodium cooling alone will not prevent deposits on the crown which can cause preignition. Such deposits do occur with unsuitable valve materials and valves with sodium in the stem only but are not known (at least to the author) where a hollow head is used.

Preignition has been produced by stem-cooled valves of only 1¼-in. head diameter at 250 psi imep. This was not a random occurrence and could be repeatedly produced after only 30 hr of operation with a new valve. It was cured by change of valve-head material.

Low-cost methods of preventing scaling of the heads of solid valves appear to be in sight.

Effects:

The effects of preignition are, in the main, due to excessive temperatures of the cylinder unit. Effects on the induction system may, however, be either thermal or mechanical. Catastrophic increases of cylinder temperature are a very usual result of preignition at high power output levels. Lacking other means, these temperature effects can usually be detected early enough to avoid damage if a spark-plug thermocouple gasket is used and closely observed.

a. Piston. The piston is usually the major item

of damage. Blowing holes in the crown or ring belt, burning, melting, or seizure are usual effects. The aircooled engine is more prone to seizure than is the water-cooled type. Burning may not lead to either seizure or melting and usually starts with pitting, which progresses until the surface resembles that of a piece of foundry or blast furnace coke. (Such pitting in aircraft engines can occur as a result of prolonged knocking, which never goes into runaway preignition.)

b. Cylinder heads. Cracking of both aircooled and water-cooled cylinder heads is a common product of preignition.

c. Valves. Stretching of exhaust valves is a common result and not unknown in intake valves. Burning and warping of exhaust valves are often found.

d. Spark plugs. Spark-plug damage is extremely likely. Some types of aviation plugs will stand repeated mild preignition (the author knows of one aviation plug which stood about 1000 mild attacks of preignition without changing its heat rating). In general, it is quite a trick to rate a spark plug for preignition without damage. Avoidance of damage is desirable in order to be able to obtain check ratings. Melting of ceramic or mica cores is by no means an unknown product of preignition.

e. Induction system. Preignition may or may not cause induction system damage. The damage may be either thermal or mechanical. In a supercharged engine having a carburetor and operating at high manifold pressure (30 psig, for example), preignition may produce backfires, which may cause parts of the induction system to depart for points unknown. In a similar engine fitted with cylinder-head fuel injection, backfires are unlikely to produce induction system damage. Induction system fires can result from backfires in carbureted engines. In aircraft engines, cases have occurred where the fire remained stationary in the supercharger volute. This, apparently, was due to the gas velocity equaling the rate of flame propagation. In any case, the pilot had no warning until the supercharger melted.[4]

Conclusions, Acknowledgments, and Apologies

This note may seem to have put altogether too much emphasis upon destructive preignition and to this the author pleads guilty, with the alibi that he has dealt mostly with what he knows or thinks he knows. It would appear, however, that the passenger-car engineer is closer to destructive preignition than he probably realizes. Preignition is distinctly a disease of high engine speed, and high volumetric efficiency at high speed aggravates the disease. The current tendency to increase the high-speed volumetric efficiency of passenger-car engines with four barrel carburetors, and so on, will tend to promote destructive preignition. Such passenger-car engines still have to perform creditably in traffic and spark plugs consequently have to be reasonably hot. The plug that is adequate for traffic driving may be quite inadequate for high speeds on the open road. In this connection, the author is unable to visualize the American driver following the practice of the European owner of a hot-stuff sports car who will willingly drive in traffic with "soft" plugs and change to "hard" plugs for the open road.

This note is almost entirely written from memory but draws freely upon the literature and the experiences of others. Since the author has not the energy necessary to search out old references in the literature, he concluded to give no references at all, hence the lack of a bibliography, which might have been quite extensive. It is trusted that this explanation may satisfy those authors who may feel slighted.

In conclusion, the author almost is tempted to blame the defects of this note on C. J. Livingstone who had the bad judgment to invite the author to write it.

Note About Appendix

The Rolls-Royce bulletin referred to in the text has been reproduced since it appears to be of considerable interest. It was originally published in April, 1918, and is reproduced with the permission of Rolls-Royce Ltd.

Explanatory notes appear to be desirable in regard to F.A.S. and "quick step" valves, mentioned under "Effect of Preignition." F.A.S. is presumably the high-speed tool steel then used for valves and containing 12% tungsten and 4% chromium; "quick step" is the Brown Bayley Quick Step steel containing about 7% chromium and about 1% silicon.

APPENDIX

Notes on Trouble Caused by the Burning of Exhaust Valves

As a result of trouble having been reported on several occasions by the Services through the burning of exhaust valves of aero engines, Messrs. Rolls-Royce Limited have carried out a series of experiments to determine the cause and to obviate, if possible, the recurrence of this trouble. The following notes are based on their report.

A particular instance of this trouble having been experienced on an aero engine after a few hours' run was brought to the notice of Messrs. Rolls-Royce Limited, and a thorough investigation was carried out by them without the engine having been interfered with in any way. It was ascertained that the seating of the valve in the cylinder was true, the tappet clearance was correct, and that the analysis of the steel was satisfactory. Rolls-Royce Limited, however, claim to have proved conclusively that the valves had burnt through preignition arising from the ignition plug.

The firm hitherto have submitted their engines

[4] This case may not have been caused by preignition but may have resulted from backfiring caused by bad distribution. It illustrates, however, a danger that can arise from backfires resulting from preignition.

periodically to rigorous tests of long duration on their testing tackle, and in all these tests no exhaust valves have burnt. They had assumed that the conditions of these tests were more adverse than Service conditions, and therefore were rather at a loss to understand the occurrence of this trouble under normal flying conditions. They have also examined repair engines that have run in Service up to 200 hr without the valves having been changed or their condition impaired.

A number of further tests having been carried out specially by the firm, they believe that the following information resulting from these investigations will prove of value and eliminate mistaken impressions at present held as to the causes of exhaust valves burning.

Temperature of Exhaust Valves – It was found that the temperature of an exhaust valve in the hottest portions is 700 to 750 C, and that the temperature of the exhaust gases in the exhaust port is approximately 700 C.

Effect of Different Mixtures on Temperature – It was found that the valve is hottest when the cylinder is working with the most efficient mixture: that is to say, when the maximum power is obtained with the minimum amount of petrol. Whether the mixture be weakened or strengthened from this point the temperature is immediately reduced. When the mixture is weakened the power immediately drops; but the mixture can be strengthened considerably without any loss of power. The general impression that weak mixtures cause the valves to get very hot was not borne out by these experiments.

It was apparent that the effect the mixture has on valves burning is controlled by the amount of free oxygen in the exhaust. With a correct mixture there is a certain percentage of free oxygen in the exhaust, whereas with a slightly stronger mixture this does not occur.

Chromium and Tungsten Steels – Chromium steel possesses the advantage over tungsten steel in that it does not oxidize at ordinary working temperatures. The disadvantage in the use of tungsten steel is that in ordinary conditions the valves scale and gradually get thinner. Once the valve reaches the temperature at which burning commences, however, the difference between tungsten and chromium steels is not appreciable.

Effect of Various Tappet Clearances – The tappet clearance is, of course, directly affected by the elongation of the valve at normal working temperatures. Tests carried out showed that this elongation is 0.014 to 0.016 in.

A set of valves were run with no clearance at all in the tappets when cold, so that at normal temperatures the exhaust valves did not seat by 0.015 in. The engine ran for 25 hr and the valves were in perfect condition when the test was finished. It was noticeable while the engine was running that the valves were the same heat all over, whereas in normal working conditions they are cooled on the seating where it comes in contact with the cylinder.

It was found that it was possible to run with the valves not seating by as much as 0.008 in. and still obtain a cooled ring round the outer ridge of the valves. It is assumed that in this case the valve is cooled by the thin volume of gas passing around the valve at a considerably reduced velocity and itself being cooled by contact with the seating of the cylinder.

Investigation showed that with no tappet clearance in the exhaust valves when cold the power was reduced by 1.5%, and further, that it was possible to run with only 0.005 in. clearance with no loss of power.

In carrying the tests still further, the tappets were adjusted so that the exhaust valves could never seat when cold by 0.01 in. As a result the power was reduced by as much as 15% and the exhaust valves became excessively hot.

In these conditions it was distinctly noticeable that the valves were hottest on the outer ring and cooler towards the stem; this, of course, being the converse of results with correct tappet adjustment.

This test also revealed another interesting point in that the valve was found to be much hotter on the sector towards the top center of the cylinder. This is of particular interest because when burning occurs in an exhaust valve it has apparently never been appreciated why a bite is taken out of one portion, leaving the rest of the seating in fairly good condition. During these experiments it was noticed that this bite always occurs on the sector just mentioned towards the top center where the valve gets hottest. In conjunction with this it was also found that if a valve is turning round while the engine is running it takes a very much longer time to start to burn. Immediately burning commences the valve ceases to turn.

Distorted Valve Seatings – To reproduce the conditions of an engine running with distorted valve seatings, the valves were filed away on the angle of the seating. Two were made so that a feeler 0.005 in. thick could be inserted between the valve and the cylinder seating; another two were made with a clearance of 0.008 in. and a further two with 0.012 in. clearance. The engine fitted with these valves ran for 15 hr. On the completion of the run the valves were examined and a slight burning effect was noticeable where the valves were not bedded, but the results were nothing like those indicated in the reports received from the Services. The firm considers, therefore, that the slight distortion that may occur in valve seats is not the cause of the valves burning.

Effect of Preignition – It was ascertained during the investigation that with a sparking plug which preignites it was possible to burn either an F.A.S. or a "quick step" valve in from six to eight minutes, and a number of valves were actually burnt out in this time during the tests.

An attempt was made on a 6-cyl engine fitted

with "Eagle" cylinders to imitate the conditions of a preigniting sparking plug. For this purpose one magneto was correctly timed and the second one connected up in such a way that it could be fired in any position up the stroke, thus enabling the operator to produce preignition at will. It was found, however, that the desired conditions could not be produced in this way, the actual conditions resulting being quite different to those where preignition is due to an overheated plug. Preignition was obtained, which considerably reduced the power, without, however, producing the excessive heat.

A further test was made by using a coil and having a continuous spark on the plug, in the expectation that this arrangement might reproduce the effect of an overheated plug, but again it was found that it was not possible to reproduce the actual conditions arising from an overheated plug.

A sparking plug which preignites under ordinary running conditions causes the exhaust valve to run at an incandescent heat, at which it is amazing that the valve stands up at all.

It was found that preignitions are greatly influenced by the amount of lubricating oil in the combustion chamber.

The firm claims that a case of preignition in an overoiled cylinder has never been known, and it was proved by actual experiments that if a plug commences to preignite, this can be remedied immediately by syringing a little oil into the air intake.

It was also proved, by watching the temperature of the exhaust valve, that it is possible to inject a certain amount of oil without affecting the horsepower of the engine, but that does reduce the temperature of the valve.

Effect of Faulty Water Circulation – It was found in the course of the tests that the water circulation has a very big effect on the overheating of exhaust valves. It was noticeable that if the water circulation is faulty the first part to be affected is the exhaust valve. By watching an exhaust valve when the engine is running it can be clearly observed that as the circulation fails the valve gets hotter and even the exhaust bend and guide get to a dull red heat, although there may be still a certain amount of water circulating.

If the failure of the water circulation causes the sparking plug to preignite, which is a natural consequence, the valve is burnt in a few minutes.

Summary – From the results of the above investigations and from the information obtained from the Services, Messrs. Rolls-Royce Limited consider that the burning of exhaust valves is due to two causes:

1. Failure of water circulation.
2. Preignition.

Of these it is thought that No. 1 is more frequently the cause of the trouble, in addition to which it may set up condition No. 2.

The firm base their reasons for this opinion on the fact that complaints of burnt valves are always more frequent in the summer months, and that the squadrons that have most trouble are those in which it is known that the water cooling of the engines is not very good.

They do not remember hearing at any time of a valve being burnt on an F.E.2.D. machine, although these particular machines were fitted with an Eagle Series 1 engine having very thin valves made of tungsten steel which, in the light of the firm's experiments, should have given more trouble than later series. At a later date Eagle engines of a more recent series were fitted into the above machine, but again no trouble was experienced in the way of burnt valves.

The firm's opinion is that, when the water circulation tends to fail the water commences to boil and, consequently, water is lost through priming and evaporation, a trouble that it has not been possible so far to obviate. As the engine continues to run the conditions get worse, and in the hottest part, namely, that portion of the water jacket round the exhaust valve, steam is generated, which tends to force the water out in the front, and consequently exerts a pressure against the water pump, which still further reduces the flow. By this time preignition starts, the power drops, and the valves commence to burn.

Messrs. Rolls-Royce Limited affirm that preignition should not occur in Service if reasonable care is taken. The chief reason for its occurrence is, in their opinion, that the engine is run with the oil too cold, which reduces the quantity that escapes from the bearings, and so in turn the quantity that reaches the pistons. As a result, conditions are set up that considerably increase the tendency to preignite. The firm fully realize, of course, that occasionally a faulty sparking plug is found which preignites and cannot be controlled.

Messrs. Rolls-Royce Limited conclude, therefore, by pointing out that at present the exhaust valve acts as a "safety valve" for overheating, and that it would be very much more disastrous if, for instance, the piston failed first. A burnt exhaust valve does not prevent the pilot getting home and the part is easily replaced; they, therefore, think that when overheating takes place the exhaust valve is perhaps the best part on which failure should occur.

They state that very encouraging results have been obtained by increasing the flow of water per minute through the cylinder jackets. Further experiments in this particular direction are now being carried out by the firm. The firm state that radiators and radiator piping offer by far the greater resistance to the flow of water, and any resistance in these should be very carefully avoided. It is quite possible that any foreign matter, other than pure water, may tend to increase the resistance through the radiator and hence cause the water to boil through reduced circulation. The priming which naturally follows would soon complete the failure of the cooling system.

An Investigation of

ALTHOUGH increases in engine compression ratios have enabled the more efficient utilization of fuels, these benefits are accompanied by an increase in the tendency for preignition to occur during road operation. One cause of preignition in modern gasoline engines is the presence of combustion-chamber deposits, which, under certain conditions, can become hot enough to ignite the fuel-air mixture prior to ignition by the spark. Early ignition results in abnormally high rates of pressure rise in the cylinder, which often are accompanied by a loud noise, and which, if severe enough, can cause actual destruction of engine parts. Preignition, therefore, may become a very real obstacle governing the performance of fuels in engines, approaching or exceeding knock in importance.

An extensive study of preignition is being car-

COMBUSTION-chamber deposits are the primary cause of preignition. This preignition may be silent or it may be manifested by a variety of engine noises ranging from light pings to violent explosions.

Its occurrence, according to these authors, is favored by changes in engine operating conditions that result in the development of higher pressures and temperatures during compression, or that cause an increase in the concentration of oxygen available to support the combustion of carbonaceous material.

Thus, increased compression ratio, supercharging, operation with retarded spark timing, and the combustion of lean mixtures are likely to increase the tendency for the occurrence of preignition.

Some hydrocarbons have a greater tendency to be ignited by hot deposit particles than others. Preflame reactions of the fuel tend to sensitize it to ignition. Under certain conditions, the effect of these preflame reactions can be minimized by tetraethyl lead.

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Preignition in Engines

A. O. Melby, D. R. Diggs, and B. M. Sturgis,
E. I. du Pont de Nemours & Co., Inc.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1953.

ried out at the du Pont Petroleum Laboratory. This paper will discuss a part of this work, which has been concerned with an investigation of the nature and causes of preignition in engines.

Forms of Preignition

The abnormal combustion phenomena that meet the definition of preignition may have several different physical manifestations. Although these manifestations have been recognized for some time, the exact nature of the changes in the combustion process which accompany them are not yet understood completely. In order to clarify the nature of the problem, experiments were conducted to learn more about the fundamentals of preignition induced by the presence of combustion-chamber deposits.

For this work a single-cylinder L-head CFR engine equipped with 17 spark plugs was used.[1] The location of these spark plugs in the combustion chamber is shown in Fig. 1. Only the spark plug in the normal location (number 1 in Fig. 1) served as an ignition source, while each of the remaining 16 spark plugs was connected through an electronic circuit to a small neon bulb, which would light when the air gap between the electrodes was ionized by the passage of a flame. Two additional neon bulbs were provided to serve as an indication of the time of spark ignition and to furnish a time calibration. All the neon bulbs were arranged in a horizontal line and the time sequence of illumination of the bulbs was determined by high-speed streak photography. These instrumentation techniques are essentially the same as those originally described by Schnauffer.[2]

High-speed strip film records were obtained during operation of the engine with both normal and abnormal combustion. A reproduction of a typical ionization pattern is shown in Fig. 2. Each line represents one of the neon bulbs, which is identified by the number shown at the right end of each line. The numbers refer to the individual ionization gaps, which are located as shown in Fig. 1. Fig. 2 represents normal combustion, and the progress of the flame front is indicated by the distance from the beginning of the line made by the ignition bulb to the beginning of the line made by the bulb connected to any particular ionization gap. Abnormal combustion is indicated by a significant departure from the normal pattern shown in Fig. 2, and preignition would be indicated by the appearance of any lines ahead of the ignition line. From such a record the time at which preignition occurred and the region in the combustion chamber where it began could be easily ascertained.

In addition to the ionization gaps, the engine was equipped with a rate of change of pressure pickup, the signal from which was observed on an oscilloscope screen. The rate of change of pressure pattern was very helpful since it afforded an instantaneous indication of preignition.

Preignition was induced by the presence of combustion-chamber deposits, which were accumulated over a number of hours of light-duty operation. During all the tests in which abnormal combustion was being studied, simultaneous note was made of the engine "noise." From this study four distinct types of preignition have been identified. These have been designated as silent

[1] See SAE Transactions, Vol. 61, 1953, pp. 402-408: "Effect of Combustion Time on Knock in Spark-Ignition Engine," by D. R. Diggs.

[2] See SAE Transactions, Vol. 29, 1934, pp. 17-24: "Engine-Cylinder Flame-Propagation Studied by New Methods," K. Schnauffer.

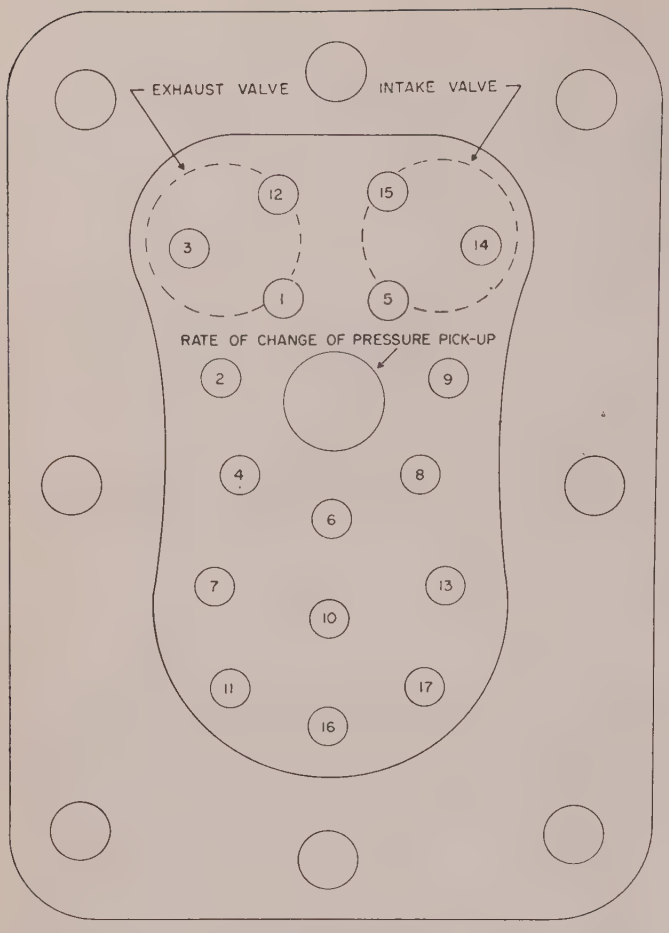

Fig. 1 – L-head CFR engine combustion chamber showing location of 17 spark plugs

preignition, steady preignition with noise, autoignition, and wild ping.

Silent Preignition – An increase in the normal rate of pressure change which occurs before ignition by the spark plug and which is not accompanied by any unusual engine noises is an indication of silent preignition. The ionization gap

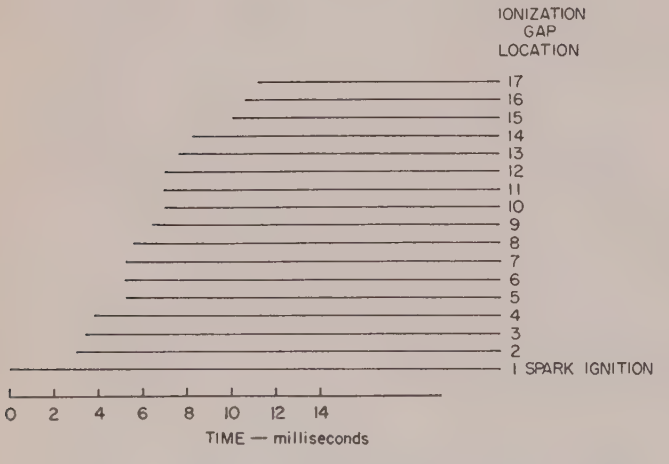

Fig. 2 – Typical ionization pattern obtained during normal combustion

records show the presence of a flame in the combustion chamber before the time of spark ignition. If the normal spark-timing is retarded sufficiently, silent preignition can be initiated early enough in the engine cycle so that combustion is essentially complete before the normal spark is passed. In this case the engine will run normally and without noise with the ignition turned off. Under these conditions, the deposits have simply replaced the spark plug as the source of ignition. The ability of the engine to run normally while silent preignition is occurring may be transient, depending on the source of preignition.

The time in the cycle at which silent preignition occurs may vary quite widely and randomly from one preigniting cycle to the next preigniting cycle, especially if a number of normal combustion cycles occur during this interval. Ionization patterns obtained during silent preignition indicated that preignition usually was initiated on the side of the combustion chamber opposite the valves, although in some instances preignition also occurred near the spark plug. There was essentially no change from cycle to cycle in the location where preignition started. This may indicate that in these instances ignition resulted from hot surface deposits rather than from detached flakes of deposit moving about in the gaseous mixture.

Steady Preignition with Noise – During steady preignition with noise, essentially the same rate of pressure change pattern is observed as when silent preignition is occurring. The engine makes abnormal noises very much like knock, but the rate of pressure change trace shows no evidence of high-frequency gas vibrations usually thought to be occurring coincident with combustion noise. This same type of preignition has also been observed in laboratory multicylinder engines equipped with both ionization detectors and knock pickups, where, when preignition was indicated by the ionization behavior, the knock indicator did not show any irregular phenomena even though a low-pitched rumbling noise was evident.

The ionization patterns obtained with the single-cylinder engine during steady preignition with noise were essentially the same as those obtained during silent preignition, and thus did not furnish any clue as to the source of the noise. A physical interpretation of this second type of preignition cannot be made as yet.

Autoignition – Preignition also is manifested by an instantaneous explosion of all or a portion of the fuel-air mixture prior to the normal spark. This explosive combustion appears to be a type of autoignition induced by the presence of heated deposit surfaces. This autoignition is accompanied by violent noise and often will persist if the ignition is turned off. If the engine is operated with retarded ignition timing, autoignition usually takes place at about the same time on each successive engine cycle. This may indicate a com-

pression-ignition phenomenon accentuated by the presence of hot deposit surfaces. Because of the instantaneous reaction, the ionization records cannot be used to establish the place in the combustion chamber where the reaction is initiated. Autoignition represents the most violent form of abnormal combustion.

Wild Ping – Wild ping, the fourth type of preignition, is characterized by sharp, intermittent noises of short duration. This type of preignition is the one most frequently observed in cars on the road. While the occurrence of these noises is not generally predictable, they most often are heard during a transition stage such as an acceleration.

A film record showing the ionization pattern for a number of successive engine cycles was obtained, during which time the engine was preigniting on every cycle. Preignition did not cause any change in the operation of the engine and noise was evident only intermittently, but the engine continued to run even though the ignition was turned off. On all but four of the engine cycles the flame progressed in an orderly fashion from its source, usually located in the end gas region, and combustion was completed before normal ignition took place. The presence of several flame fronts was indicated during some of these cycles, and rough approximations of the burning time in these instances showed more rapid combustion than would be expected from only a single flame front. On the other four engine cycles the film record showed that autoignition of the entire mixture occurred before the spark was passed.

During the time the records were being obtained wild ping was heard four times. The correspondence of the number of autoignition cycles with the number of wild pings suggests strongly that the sound of wild ping results from an autoignition of the fuel-air mixture. If autoignition occurs only irregularly the characteristic sound of wild ping will be heard, whereas autoignition on every cycle will result in the much more violent noise already discussed as the third type of preignition. The level of noise associated with wild ping or autoignition is a function of the amount of mixture involved in the rapid reaction and also perhaps of the time in the cycle at which the reaction takes place.

The type of preignition that is observed at given engine operating conditions undoubtedly depends on the preignition resistance of the fuel being used. Also, the deposits may influence the course of the combustion process in a number of different ways. It has been established that the tendency of deposits to cause preignition is intimately related to their flaking characteristics. It also has been shown[3] that deposit flakes may become hot enough to glow, and that in this state they may serve as an ignition source either while still attached to the combustion-chamber wall or while floating about in the combustion space. Visual inspection of the combustion-chamber deposits both before and after the tests described above showed that deposits which had a flaked appearance were most effective in promoting preignition, while deposits which were relatively smooth and uniform were much less prone to induce preignition. In fact, wild ping did not occur in the absence of flaking. Since the wild ping type of preignition is the one most often observed in cars on the road, and because of the evident importance of deposit flakes in promoting wild ping, the activity of deposit particles was examined in considerable detail.

Preignition Induced by Detached Deposit Particles

Deposit particles that are detached from the walls and remain in the engine for one or more cycles are likely to be more harmful as sources of preignition than deposits that remain attached to combustion-chamber surfaces. The detached particles will usually be at higher temperatures because they are in more intimate contact with the hot gases and are not cooled appreciably by heat losses to the combustion-chamber walls. Since detached particles move with the gas stream, they remain in contact with the same small portion of the combustible mixture for a longer time and can more readily raise the temperature of this portion to its ignition point.

The preignition-inducing characteristics of de-

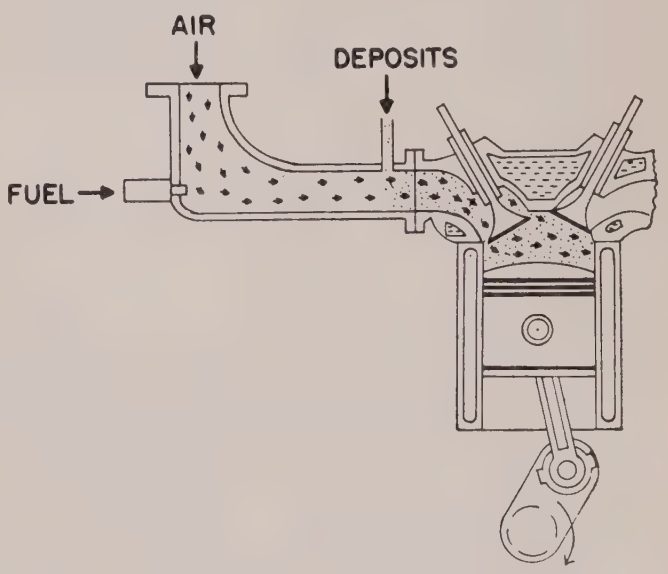

Fig. 3 – Method of particle introduction

[3] See SAE *Quarterly* Transactions, Vol. 6, 1952, pp. 724-751: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.

posit particles are difficult to study because flaking is entirely unpredictable both as to the location and time of its occurrence, and because particles of different sizes are involved. To minimize these difficulties, preignition resulting from deposit flaking was simulated by introducing weighed amounts of carefully sized deposit particles into the combustion chamber of a Waukesha single-cylinder Cooperative Oil Test (C.O.T.) engine.

Apparatus and Procedures – Particles of deposits scraped from automotive engines and mixtures of deposit components which had been prepared in the laboratory were supplied to the C.O.T. engine through a small valve that was mounted on the upper side of the inlet manifold approximately one inch from the entrance to the inlet valve port. A sketch of this installation appears in Fig. 3. Particles were fed by gravity through this valve into the inlet manifold, where they were entrained by the air stream and carried into the combustion chamber.

Effects of particles on the ignition and combustion processes were detected by observing the rate of pressure change-crank angle diagram produced on the screen of an oscilloscope by a Phillips internal pickup unit. The extent and severity of preignition were evaluated by counting the number of preignitions and estimating the crank angle at which they occurred. The counting was done by a four-decade electronic counter, which was actuated by a suitable signal from the internal pickup.

Observations and Discussion – Effects of particle composition. In initial experiments[4] it was found that severe preignition accompanied the introduction into the combustion chamber of particles consisting of carbon-lead salt mixtures and particles of deposits formed in engines that had been operated on fuels containing tetraethyl lead. Only light preignition was caused by engine deposits formed using unleaded fuels and neither pure carbon nor the lead salts themselves exhibited any appreciable tendencies to induce preignition. These observations, together with the earlier work of Campbell[5] which showed that lead salts lower the ignition temperature of carbon and promote its oxidation, suggest that deposits attain the temperatures requisite for preignition through the heat which is released by the combustion of carbonaceous material.

Factors which influence carbon combustion. The combustion of deposit carbonaceous material is influenced by the temperature of the deposit, the concentration of oxygen in the gases surrounding the deposit, and catalytic effects of other materials present in the deposit. Carbonaceous material, whether in a detached particle or in a deposit which is anchored to the combustion-chamber surface, will glow if some minimum temperature is reached, and oxidation will proceed at a rate proportional to the concentration of oxygen which is available to support combustion. It is to be expected that any change in engine operating conditions which produces an increase in combustion-chamber temperature or makes more oxygen available will favor the occurrence of preignition.

Metal salts act as catalysts for the combustion of deposit carbonaceous material by lowering the threshold ignition temperature and accelerating the oxidation once it has started. Lead salts are the principal metallic constituents of combustion-chamber deposits, and as such, are primarily responsible for promoting the burn-off of carbonaceous material. Other metal compounds, which enter the combustion chamber with the inlet air, or are formed through wear and corrosion of engine parts, also promote the combustion of carbon.

The effects of engine operating conditions. The ability of deposits to cause preignition depends not only on their chemical composition but also on their temperature and the amount of oxygen present in their environment, since these factors control the rate of oxidation of the carbonaceous material. The occurrence of preignition also is dependent upon the igniting characteristics of the fuel used in the engine. In order to investigate the interrelation of these factors, particle introduction-type experiments were conducted under a wide range of engine operating conditions. These conditions are summarized in Table 1.

For most experiments the engine was operated with spark ignition occurring at 5 deg btdc and with the electronic preignition counter adjusted to register whenever preignition occurred 5 deg or more before the normal spark or 10 deg btdc. The deposit particles which were used for studying the effects of engine operating conditions on preignition were scraped from the tops of the pistons of a 1949 passenger-car engine that had been operated in the laboratory for 200 hr. The fuel was a catalytically cracked distillate contain-

[4] See SAE *Quarterly* Transactions, Vol. 6, 1952, pp. 751-752: discussion by B. M. Sturgis.

[5] United States Patent 2,405,560 (1946) to J. M. Campbell.

Table 1 – C. O. T. Engine Operating Conditions for Studies of Deposit Preignition Harm

Fuel	Benzene
Fuel System Type	Manifold injection
Air/Fuel Ratio	Variable
Inlet Air Temperature, F	160 and 300
Manifold Air Pressure, in. Hg abs	30 to 65
Jacket Temperature, F	216
Oil Temperature, F	145
Compression Ratio	5.4/1 and 6.8/1
Engine Speed, rpm	2370
Spark Timing	Variable
Spark-Plug Type	Very cold

ing 3.0 ml of tetraethyl lead per gallon, and the lubricant, a compounded SAE 30 oil.

Compression ratio. Increases in compression ratio favor the occurrence of preignition by producing higher pressures and temperatures during compression, which decrease the ignition energy requirements of the combustible mixture[6] and increase the igniting potential of the particle. The latter effects result from an increase in both the temperature and the partial pressure of oxygen in the system.

The effect of compression ratio on the amount of preignition, caused by particles of engine deposits and of sugar carbon, is illustrated in Fig. 4. A fivefold increase in the number of preignitions induced by the engine deposits resulted when the compression ratio was increased from 5.4/1 to 6.8/1. The importance of increased compression ratio is indicated further by the behavior of carbon particles which cause very little preignition at the lower compression ratio, but appreciable preignition at the higher compression ratio.

Inlet manifold air pressure. Increasing the inlet manifold air pressure caused increases in deposit preignition harm similar to those which resulted from increased compression ratio. Fig. 5 shows that the number of preignitions caused by engine deposit particles increased linearly as the manifold air pressure was increased from 30 to 65 in. Hg abs. It is believed that this increase can be attributed to an increase in the ignitability of the charge and the ratio of oxygen to carbonaceous material.

Spark timing. Marked increases in the preignition harm of detached deposit particles occurred as spark timing was retarded. This is illustrated in Fig. 6, where the number of preignitions occurring 10 deg or more before spark are plotted against spark timing for inlet air temperatures of 160 F and 300 F.

The greater number of preignitions under retarded spark conditions may be explained by (1) an increase in the temperature of the exhaust gases, (2) an increase in the time available for preignition, and (3) the increases in mixture ignitability and particle igniting potential which result from the increases in charge density accompanying further compression.

The temperature of the gaseous products in the combustion chamber during the power and exhaust strokes determines to a large extent the temperature of all surfaces with which the fresh charge will come into contact. Thermal effects of the combustion gases are particularly important because the temperature, and hence the igniting ability, of deposits and surfaces which are not participating in chemical reactions is determined

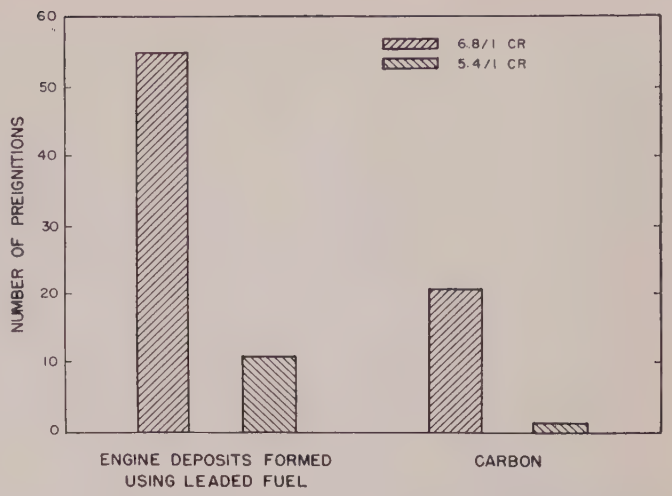

Fig. 4 – Effect of compression ratio on preignition resulting from introduction of engine deposit particles and carbon. 40 in. Hg manifold air pressure, 300 F inlet air temperature, 12.5 air/fuel ratio, 5 deg btdc spark timing

entirely by the temperature of their environment. However, the temperature of deposits which are undergoing exothermic chemical reactions, such as are involved in the burn-off of carbonaceous material, is likely to be influenced by the combustion gases to a lesser extent.

The greater time available for particle-caused preignition as the spark timing was retarded was probably more important in promoting preignition than the higher temperature level of the exhaust gases. This longer contact time permitted the transfer of a greater amount of energy from particles to the combustible mixture, thus increasing the probability of preignition. In addition, the ignitability of the mixture and the igniting potential of the chemically active particles become greater with each additional degree of crank

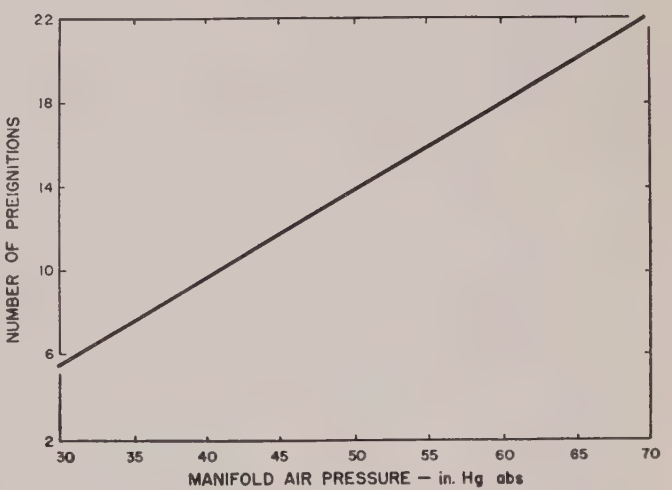

Fig. 5 – Effect of manifold air pressure on preignition resulting from introduction of engine deposit particles. 300 F inlet air temperature, 12.5 air/fuel ratio, 5 deg btdc spark timing 5.4/1 compression ratio

[6] See "Combustion, Flames, and Explosions of Gases," by B. Lewis and G. von Elbe. Pub. by Academic Press, Inc., New York, 1951.

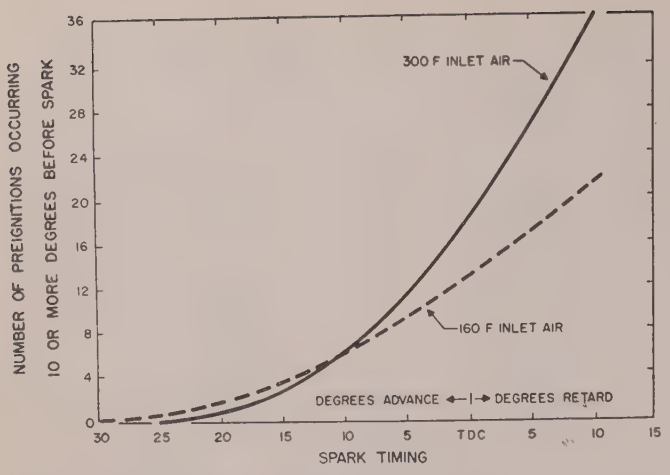

Fig. 6 – Effect of spark timing and inlet air temperature on preignition resulting from introduction of engine deposit particles. 40 in. Hg manifold air pressure 12.5 air/fuel ratio, 5.4/1 compression ratio

travel during compression because of the increase in temperature and pressure. The increased ignitability of the mixture is due in part to the occurrence of fuel oxidation reactions which sensitize it to ignition.

Inlet air temperature. A moderate increase in the amount of preignition caused by particles was found to occur when the temperature of the inlet air was raised from 160 F to 300 F, as shown in Fig. 6. This effect was more pronounced with late spark timing than with more advanced ignition. The effects of changes in air temperature are less marked than might be expected. Increasing the inlet air temperature favors the occurrence of preignition by raising the operating temperature level, but at the same time decreases the charge density, which tends to reduce deposit preignition harm.

Air/fuel ratio. The introduction of engine deposit particles did not cause preignition under extremely rich-mixture operating conditions, but as air/fuel ratio was increased, preignition occurred in increasing amounts. This effect is shown in Fig. 7.

Operation at lean mixtures increases the oxygen concentration in both the unburned and burned gases, increases the temperature level of the combustion chamber, and decreases the ignitability of the mixture. The increase in oxygen concentration is particularly important because oxygen must be available if carbon combustion, much of which is initiated at the time of charge combustion, is to be maintained during the power and exhaust strokes to the extent necessary to provide the temperatures requisite for preignition on subsequent compression strokes. During the intake and compression strokes the effects of increasing the air/fuel ratio on oxygen concentration are very small and probably negligible. After combustion and throughout the expansion and exhaust strokes, however, the oxygen concentration is much higher under lean- than under rich-mixture conditions since more of the oxygen remains after the combustion. Because of this, carbon burn-off is sustained and probably even promoted.

Particle-Induced Preignition on the Road – To determine the effectiveness of particles as sources of preignition in automobiles, four different makes of recent-model passenger cars were equipped with apparatus which permitted the introduction of particles into one cylinder during road operation. The induction system of the car was modified so that a small quantity of powdered deposit could be inducted just ahead of the intake valve at the beginning of full-throttle acceleration to a road speed of 55 mph. During this acceleration, any preignition that occurred was detected by ear. The combustion chambers of the cars were cleaned thoroughly prior to the road tests, and the spark-ignition timing was adjusted to the manufacturer's specifications. Primary reference fuel blends having antiknock qualities slightly greater than necessary for the suppression of knock were used as test fuels. Neither knock nor preignition was observed at any time when deposit particles were not introduced. Effects resulting from the introduction of 1.0 gram of 0.006-0.012-in. particles are summarized in Table 2.

Engine noises resulting from the introduction of both leaded fuel deposit particles and carbon-lead oxide particles were essentially the same as the wild ping type of preignition noise, which often occurs during road accelerations. These observations suggest, therefore, that detached

Table 2 – Preignition from Particle Introduction during Road Acceleration

Particles	Estimated Number of Preignitions	Type of Noise
Leaded fuel deposits	6	Moderately sharp cracks
Unleaded fuel deposits	2	Dull thuds
Sugar carbon + PbO	21	Sharp, violent cracks
Sugar carbon + $PbCl_2$	2	Dull thuds
Sugar carbon	1	Dull thuds

Table 3 – Preignition Resistance of Fuels

	Number of Preignitions
Isooctane	20
Mixed Xylenes	24
Toluene	26
Benzene	55

deposit particles may be one cause of wild ping in cars on the road.

Preignition Resistance of Fuels

Although the occurrence of preignition is controlled to a large extent by the igniting properties of the deposits, it is affected also by the ignition characteristics of the fuels being used. The introduction of engine deposit particles into a C.O.T. engine which was operated on a number of different hydrocarbons showed that fuels differed in their resistance to ignition. Benzene preignited readily, while isooctane, xylene, and toluene were much more resistant to preignition. These data are summarized in Table 3.

These results are in accordance with those of Calcote and associates,[7] who showed that the amount of energy required for the ignition of isooctane was approximately three times as great as that required for the ignition of benzene.

Preflame Reactions and Preignition – Many hydrocarbons undergo chemical reactions before the time of spark ignition. These reactions lead to the formation of peroxides and other reactive products, which might be expected to condition or sensitize the combustible mixture and increase its tendency to ignite.[8] Evidence of this sensitizing effect is presented in Table 4, where it is shown that an 88-octane-number blend of isooctane and n-heptane is far less resistant to deposit-induced preignition than isooctane.

n-Heptane is known to undergo extensive preflame reactions early in the compression stroke, whereas isooctane, especially under the comparatively mild C.O.T. engine operating conditions used for these tests, does not.

The addition of 3.0 ml of tetraethyl lead per gallon to the 88-octane-number blend appeared to eliminate the preignition-sensitizing effect of the preflame reaction products of n-heptane. This observation, which is shown in Table 4, suggests that fuel additives may be a means of increasing the preignition resistance of some hydrocarbons. It should not be inferred from these data that a general relationship exists between the antiknock and preignition-suppressing actions of tetraethyl lead. The time-temperature characteristics of hydrocarbon oxidation and the igniting ability of the preignition source are not necessarily related in such a manner that fuel knocking and preignition qualities are influenced to similar degrees by tetraethyl lead.

Summary

Combustion-chamber deposits are the primary cause of preignition in modern automotive engines. Deposit-induced preignition may be silent or it may be manifested by a variety of engine noises ranging from light pings to violent explosions. The ability of deposits to induce preignition arises from localized high temperatures, which are developed through the combustion of carbonaceous material in the deposit structure. The burn-off of carbonaceous material is promoted by the salts of lead and other metals and is influenced markedly by engine conditions. Changes in engine operating conditions which result in the development of higher pressures and temperatures during compression, or which cause an increase in the concentration of oxygen available to support the combustion of carbonaceous material, favor the occurrence of preignition. Thus, increased compression ratio, supercharging, operation with retarded spark timing, and the combustion of lean mixtures are likely to increase the tendency for the occurrence of preignition.

Some hydrocarbons have a greater tendency to be ignited by hot deposit particles than others. Preflame reactions of the fuel tend to sensitize it to ignition. Under certain conditions, the effect of these preflame reactions can be minimized by tetraethyl lead.

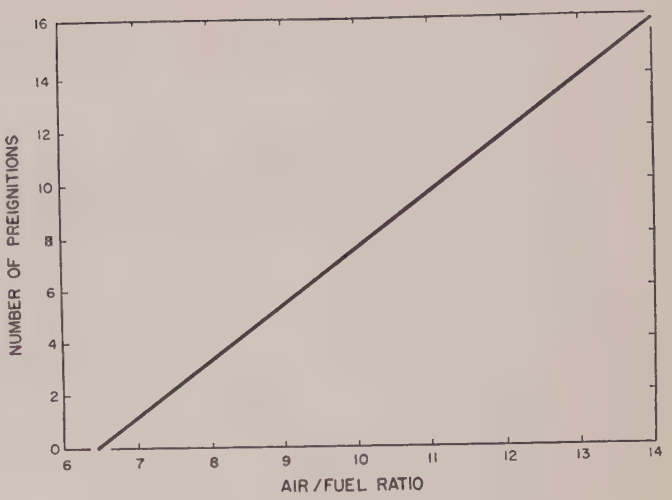

Fig. 7 – Effect of air/fuel ratio on preignition resulting from introduction of engine deposit particles. 40 in. Hg manifold air pressure, 300 F inlet air temperature, 5 deg btdc spark timing, 5.4/1 compression ratio

Table 4 – Effect of Preflame Reactions and of Tetraethyl Lead on Preignition

	Number of Preignitions
Isooctane	2
88-Octane-Number Primary Reference Fuel	22
88-Octane-Number Primary Reference Fuel + 3.0 Ml Tel/Gal	1

[7] See *Industrial and Engineering Chemistry*, Vol. 44, November, 1952, pp. 2656-2662: "Spark Ignition – Effect of Molecular Structure," by H. F. Calcote, C. A. Gregory, Jr., C. M. Barnett, and R. B. Gilmer.

[8] "Preflame Oxidation of Hydrocarbons in a Motored Engine," by A. J. Pahnke, P. M. Cohen, and B. M. Sturgis. Presented at the 124th Annual American Chemical Society Meeting, Chicago, September, 1953.

Deposit-Induced Ignition—

COMBUSTION-chamber deposits are responsible for two problems which hinder the utilization of higher engine compression ratios as a means of improving engine efficiency. One of the undesirable effects of deposits is their tendency to increase the fuel antiknock requirement of the engine for the suppression of ordinary knock. From the engine builder's viewpoint, deposits are responsible for lower compression ratios than would otherwise be used, while to the refiner they represent wasted octane numbers. Many of the factors affecting increase in ordinary knock due to deposits have been covered in a number of papers. Not so well understood is the problem of

DEPOSIT-induced ignition (the erratic ignition of the fuel-air mixture by combustion-chamber deposits) is one of the problems hindering the development of higher compression, more efficient engines. Deposit-induced ignition results in uncontrolled combustion, which often is followed by knock. In some modern engines, the suppression of knock originating through this mechanism may require higher fuel antiknock quality than that required to suppress ordinary knock.

Fuel composition and volatility have been found to affect the amount of deposit ignition. Reduction in fuel end point reduces deposit ignition. Among individual leaded hydrocarbons, aromatics produce by far the most deposit ignition, but the differences among full-boiling gasoline stocks of similar volatility do not appear to be related to their hydrocarbon-type proportions. Engine operating conditions favorable to carbon formation tend to increase deposit ignition and magnify differences among fuels.

Both the characteristics and the amount of lubricating oil which enters the combustion chamber have a marked influence on deposit ignition. One factor which contributes to differences among oils is volatilty; lower-boiling oils reduce deposit ignition.

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Evaluation in a Laboratory Engine

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This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1953

deposit-induced ignition, which is the tendency of deposits to ignite the fuel-air mixture erratically and to produce uncontrolled combustion noticeable as autoignition, wild ping, preignition, and engine roughness. In this paper, no attempt has been made to differentiate among these various effects of uncontrolled combustion. Instead, the work reported herein has been directed toward the measurement and study of deposit-induced ignition itself, the basic cause of the trouble.

Deposit-induced ignition (hereafter deposit ignition) presents problems which are quite different from those of ordinary knock. A fuel of sufficient antiknock quality can eliminate ordinary knock and also the audible effects of deposit ignition. However, the fuel octane number required to eliminate the audible effects of deposit ignition often may be higher than that required to eliminate ordinary knock. Furthermore, even when the audible effects of deposit ignition are suppressed by this means, inaudible effects such as engine roughness may still remain because of high rates of pressure rise caused by multiple, deposit-ignited flame fronts. Because of this, deposit ignition may prevent the engine designer from controlling combustion in a manner to take maximum advantage of high-octane-number fuels. Another practical aspect of the deposit-ignition problem is that generally neither the audible nor inaudible effects can be eliminated by retarding the ignition timing, although ordinary knock can readily be prevented in this manner.

To date, most methods developed for evaluating deposit ignition have depended on observations of its audible secondary effects. To provide a means for measuring more directly the tendencies of deposits to ignite the fuel-air mixture, special instrumentation was developed by the Ethyl Laboratories several years ago. This paper describes both the use of this instrumentation in a laboratory engine test and the effects of several fuel and oil factors that have been studied.

Method of Approach

Deposit ignition depends on two factors: (1) the tendency of the oil-fuel combination to form deposits having the ability to ignite the charge, and (2) the ease with which the fuel may be ignited by these deposits. The resistance of various types of fuels to hot surface ignition has been investigated but, while some differences have been found among fuels under certain specific engine conditions, this approach neglects the important deposit-forming characteristics of the fuel and oil. A more practical approach would appear to be one which would permit a simultaneous evaluation of both fuel and deposit characteristics.

Since deposit ignition may or may not produce audible effects, as will be discussed later, the audible methods used to study ordinary knock are not entirely adequate. Because of this, it was necessary to develop a method for detecting the occurrence of ignition of the fuel-air mixture when caused by the deposits. The use of ionization gaps to study flame propagation in a combustion chamber is well known. This principle, which makes use of the increase in electrical conductivity accompanying burning of the fuel-air mixture in the vicinity of the ionization gap, appeared to be adaptable to the direct measurement of deposit ignition. Exploratory studies were, therefore, made to determine the utility of this principle.

Preliminary Studies – Initially, studies were made in an L-head single-cylinder engine, in which 24 ionization gaps were spaced within the

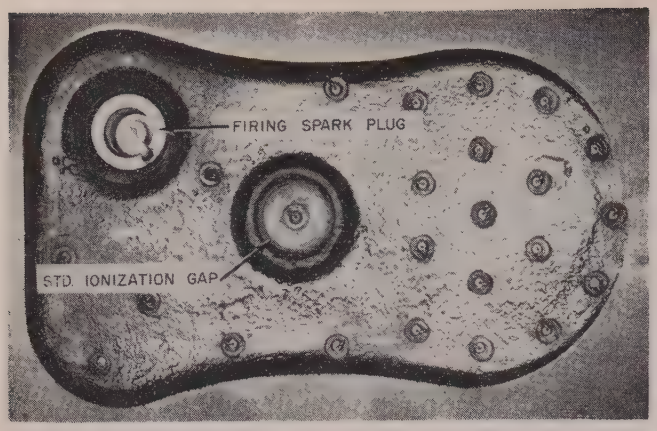

Fig. 1 – Test engine cylinder head equipped with 24 ionization gaps

combustion chamber. These gaps were connected to an electronic apparatus for determining the point in the engine cycle at which flame appeared at any given gap. It was found that during normal combustion the flame appeared at each gap within a predictable time range following normal spark ignition in each engine cycle. With deposits in the combustion chamber, however, flame occasionally was detected at times well in advance of its normal occurrence, and was found to be due to ignition by deposits. A survey of deposit-ignited combustion at the various gaps indicated that deposit ignition originated predominately in the area above the piston, although occasional instances of deposit ignition were detected in all parts of the combustion chamber.

It was also found that a single ionization gap centrally located in the combustion chamber would give a reliable indication of the amount of deposit ignition which occurred over a period of engine operation, and would thereby provide a basis for comparing different oil-fuel-additive combinations. The location of the single gap in the combustion chamber was not critical but it was found desirable to locate it at a distance from the spark plug so that some time would elapse between spark ignition and the arrival of the resultant normal flame front at the gap. This time interval permitted detection of deposit ignition by the ionization gap, not only during the period before spark ignition, but also during the period after spark ignition before the normal flame front passed the ionization gap. For simplicity, a single ionization gap in the combustion chamber was used in the rest of the work described in this paper. Fig. 1 is a photograph of a cylinder head which was used, and shows the location of the firing spark plug, the standard single ionization gap, and the other small ionization gaps which were used to detect deposit ignition during the preliminary studies.

When the engine was operated at 900 rpm with spark ignition at top dead center, the normal flame front arrived at the standard single ionization gap 15 to 18 crank-angle deg after top dead center. When deposit ignition occurred, flame occasionally arrived at the ionization gap as early as 21 deg before top dead center, or 36 to 39 deg ahead of its normal occurrence at this point. However, deposit ignition usually occurred later in the cycle, and the intensity of the audible knock resulting from it was found to depend on how early in the cycle deposit ignition occurred. Fig. 2 shows that when deposit ignition was late in the cycle it was noticeable only as inaudible engine roughness, but as its occurrence became earlier in the cycle audible knock occurred and became progressively more intense.

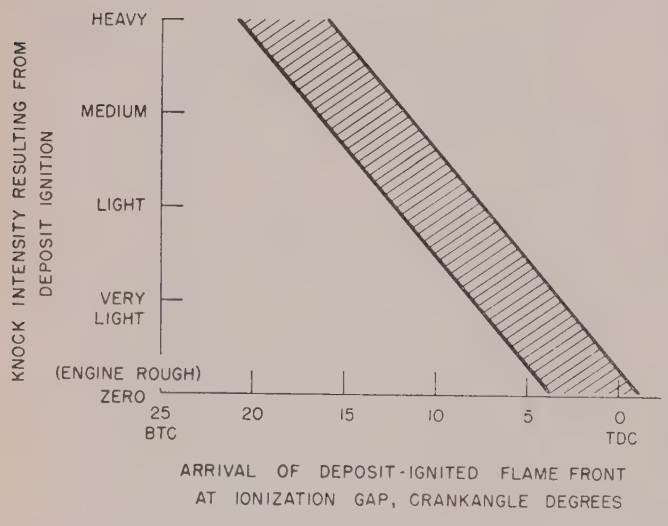

Fig. 2 – Relation of knock intensity resulting from deposit ignition to time of arrival of flame front at ionization gap. Spark ignition at top dead center. Spark-ignited flame front arrives at ionization gap at 15-18 deg after top dead center

These findings were evidence that the basic problem of deposit ignition could best be studied by the detection of early abnormal combustion independently of the audible reactions which might or might not follow. To accomplish such a study, instrumentation was developed to detect and count the instances of deposit ignition over an extended test period, so that different oil-fuel-additive combinations could be compared conveniently.

Deposit-Ignition Counter

A deposit-ignition counter was developed which, in conjunction with an ionization gap, would auto-

matically detect uncontrolled combustion arising from ignition by the deposits and would differentiate it from normal combustion. This instrument also incorporated means for recording a cumulative total of the engine cycles in which deposit ignition occurred. Differentiating between normal and uncontrolled combustion was accomplished by making the counter operate only during a selected portion of the engine cycle before the normal flame front arrived at the ionization gap. Fig. 3 illustrates this portion of the cycle.

In the previous section it was pointed out that the spark-ignited flame front arrived at the ionization gap in the test engine at 15 to 18 deg after top dead center. As shown graphically in Fig. 3, the counter was controlled to record the occurrence of flame fronts at the ionization gap during the period from 70 deg btc to 10 deg atc. Since it was rendered inoperative after 10 deg atc, the normal spark-ignited flame fronts were not counted. The 5-deg interval between cutoff of the counter and the time of arrival of the spark-ignited flame front was provided as an allowance for variations in normal flame movement which might otherwise occasionally cause an unwanted recording.

The single ionization gap cannot record late deposit ignition which causes flame fronts to arrive at the ionization gap after 10 deg atc (counter cutoff). Furthermore, while there may be several deposit-ignited flame fronts occurring in any given engine cycle, the counter cannot record more than one. However, the system does give a representative measurement of deposit ignition during a large part of that period in the engine cycle when it can occur.

The electrical circuits in the counter are illustrated schematically in Fig. 4. A positive potential of 300 v is applied through a 5-megohm resistor to the center electrode of an automotive-type spark plug, which serves only as the ionization gap in the combustion chamber. As the conductivity of the ionization gap changes, due to the presence of free ions in the flame, the voltage change at the resistor signals the arrival of the flame front. This voltage change is transmitted through the cathode follower to the gating amplifier, which provides amplification only during the portion of the engine cycle in which flame fronts are to be recorded. This on-off control of the amplifier is accomplished by a set of breaker points operated by the engine camshaft. The amplified signal from the ionization gap then passes on to the pulse generator, which in turn provides the necessary voltage to actuate the electromechanical recording counter.

The installation of this instrument on the test engine is shown in Fig. 5. The components of both the spark-plug and ionization-gap circuits are shielded so as to prevent interference between them. The breaker points controlling the counter

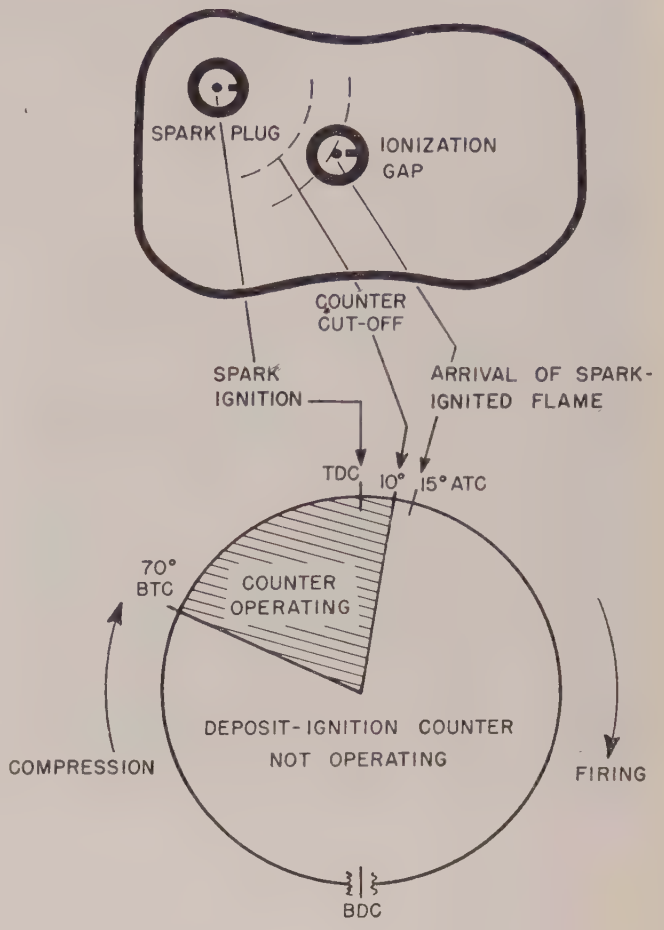

Fig. 3 – Portion of engine cycle during which deposit-ignited flame fronts are recorded. Any deposit-ignited flame front arriving at ionization gap prior to 10 deg atc (counter cutoff) is recorded by counter

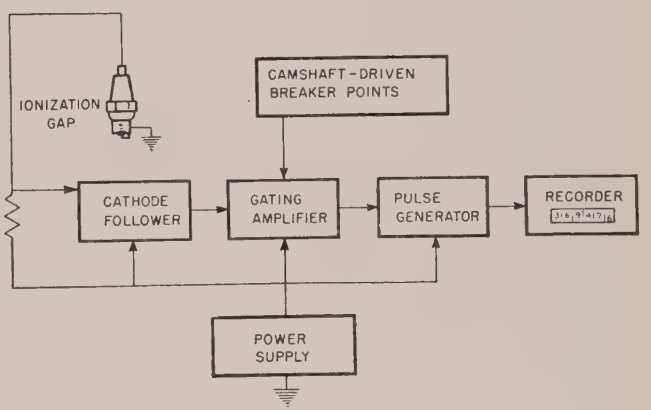

Fig. 4 – Block diagram of electrical circuits for deposit-ignition counter

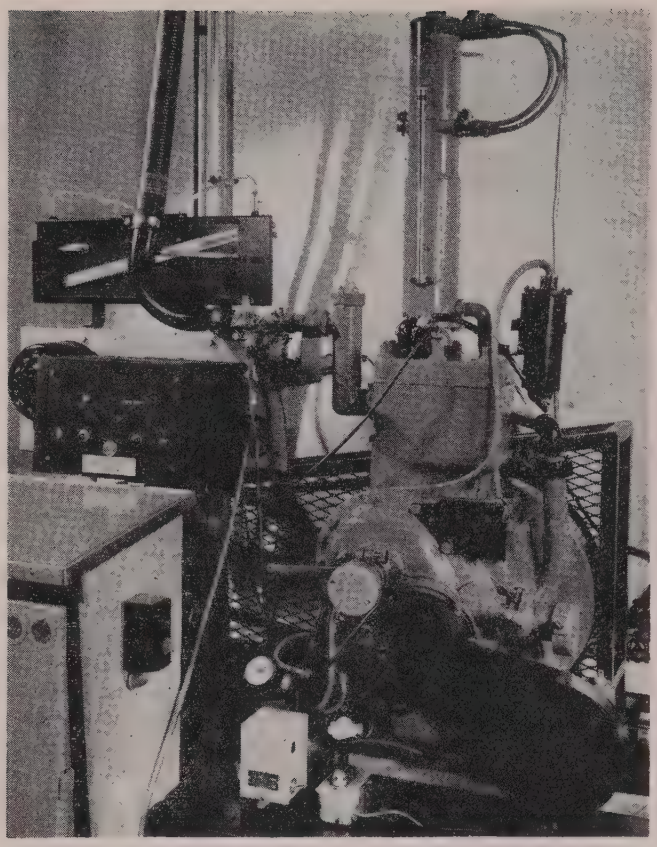

Fig. 5 – Test engine equipped with deposit-ignition counter

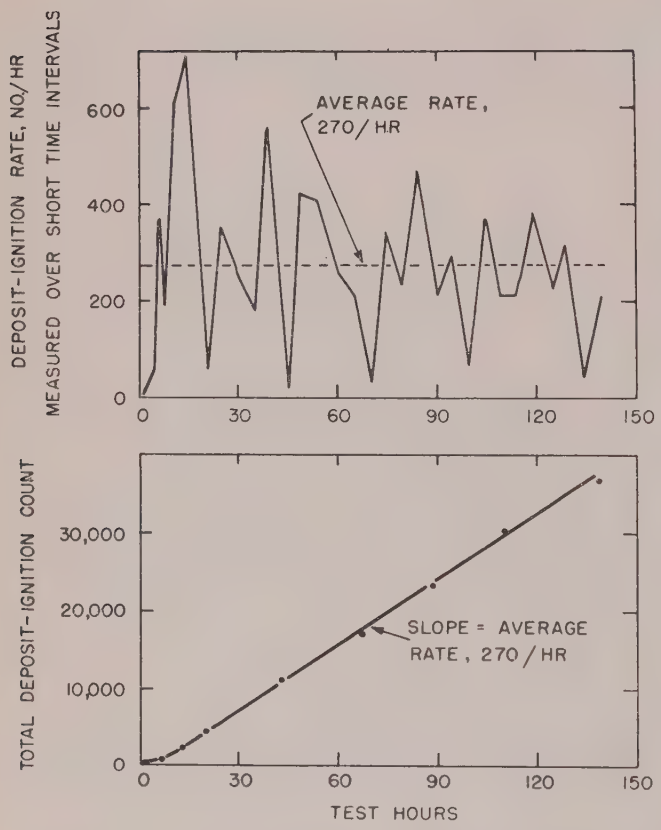

Fig. 6 – Typical results obtained with deposit-ignition counter

are shown in the lower right portion of the photograph, and are similar to the breaker points controlling the spark-ignition circuit. The deposit-ignition counter is shown in the left portion of the photograph.

Engine Test Procedure

A single-cylinder engine was used in conjunction with the deposit-ignition counter for routine evaluation of oil-fuel-additive combinations. This type of engine was most readily adaptable to the instrumentation; in addition, it permitted testing of comparatively small samples of fuels and oils under closely controlled operating conditions. The engine selected was a CFR knock-test unit in which the standard overhead-valve cylinder was replaced by an L-head cylinder.

The engine was operated under a cycling test schedule, which had been developed to evaluate combustion-chamber deposits in respect to both deposit ignition and requirement increase for ordinary knock. This cycling schedule (summarized in Table 1), consists of alternate periods of idling and full-throttle operation. The schedule starts with a clean combustion chamber and continues until the octane-number requirement for ordinary knock becomes stabilized. In order to increase the sensitivity of the test procedure, the ignition timing was retarded to top dead center. This delayed the occurrence of normal combustion and thereby provided more time for the deposits to ignite the fuel-air mixture. The use of this cycling test procedure for studying requirement increase has been discussed in a previous paper,[1] which pointed out that the procedure separated various oil-fuel-additive combinations satisfactorily with respect to passenger-car operation. Some qualitative correlation has also been found between the results obtained with the deposit-ignition counter and the audible effects of deposit ignition measured in some modern engines operated in passenger-car road service.

Fig. 6 illustrates some typical data obtained with the laboratory engine procedure. The upper portion of the figure shows the marked variation in deposit-ignition rate which is found when it is measured at intervals of 5 hr or less, and illustrates the erratic occurrence of deposit ignition. Because of this characteristic, it is desirable to plot the total cumulative count obtained over the period of the entire test, as shown in the lower portion of the figure. The average slope of this curve represents the average rate of deposit-ignitions per hour. In this typical case, the slope of the plot becomes constant after approximately the first 15 hr of deposit accumulation. This makes possible a much

[1] See SAE Transactions, Vol. 61, 1953, pp. 361-376 + (disc) 376-377: "Combustion-Chamber Deposition and Knock," by H. J. Gibson, C. A. Hall, and D. A. Hirschler.

shorter test if deposit-ignition rate is to be determined without measurement of equilibrium requirement increase.

Deposits formed during approximately 15 hr of operation in many cases can produce a rate of deposit ignition as great as that produced by larger amounts of deposits formed over a longer time. However, although the average rate of deposit ignition may not increase as deposits continue to accumulate after 15 hr of operation, the proportion of instances of deposit ignition which result in audible knock becomes greater. The upper portion of Fig. 7 shows that the rate of deposit ignition becomes high in the early stages of the test, but that very little of it produces audible knocking. In the later stages of deposit accumulation, the average rate of deposit ignition remains the same, but a much larger proportion becomes detectable by audible means. The curve showing the increase in audible deposit ignition is similar to the curve in the lower portion of the figure, which shows the increase in octane-number requirement of the engine for suppression of ordinary knock. It thus appears that, as the engine requirement increases and approaches the octane number of the fuel, the proportion of deposit ignition which is audible increases in a similar manner, although the total occurence of deposit ignition remains the same.

This would indicate that an increase in fuel antiknock quality would reduce the audible effects of deposit ignition. This is known to be true, since increasing fuel octane number is the customary method of evaluating and suppressing audible deposit ignition in passenger-car operation on the road. The data would further suggest that the total deposit-ignition rate as measured by the deposit-ignition counter would not necessarily be influenced by fuel octane number. Tests were made to investigate this possibility. Three different fuels, widely different in octane number, were prepared from blends of toluene, isooctane, and normal heptane. To maintain the same proportion of aromatics and paraffins the same amount of toluene was used in each blend, while octane number was controlled by

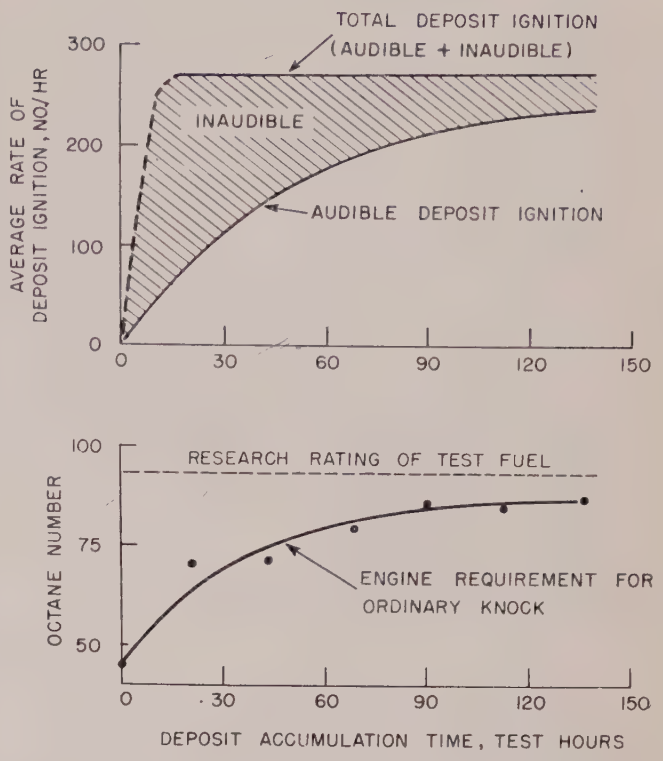

Fig. 7 – Relation of deposit accumulation time to octane requirement and to rate of deposit ignition, both audible and inaudible

Table 1 – Cycling Operating Conditions Used in the Laboratory Test Engine

	Idle	Full Throttle
Cycle Duration, sec	50	150
Speed, rpm	600	900
Fuel-Air Ratio	0.087	0.077
Ignition Timing	Tdc	
Jacket Coolant Temperature, F	150	
Oil Temperature, F	160	
Intake Air:		
Temperature, F	112	
Moisture, grains/lb	34	
Pressure	Atmospheric	
Compression Ratio	7.0/1	
Test Duration, hr	165	
Fuel Required, gal	60	

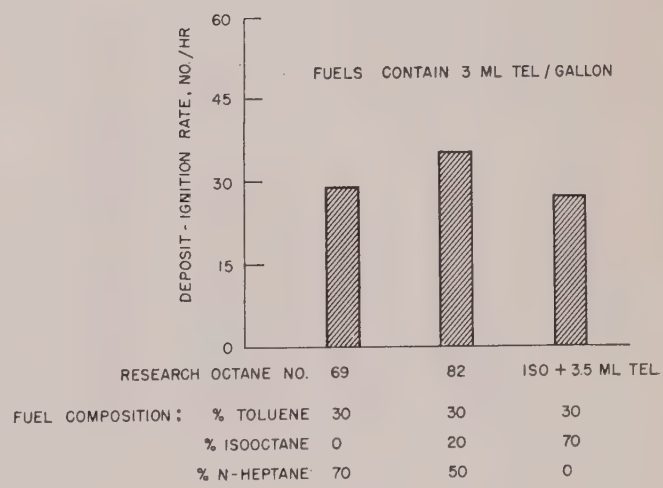
FUELS CONTAIN 3 ML TEL / GALLON
DEPOSIT - IGNITION RATE, NO./HR
60
45
30
15
0
RESEARCH OCTANE NO. 69 82 ISO + 3.5 ML TEL
FUEL COMPOSITION: % TOLUENE 30 30 30
% ISOOCTANE 0 20 70
% N-HEPTANE 70 50 0

Fig. 8 – Effect of fuel octane number on deposit ignition

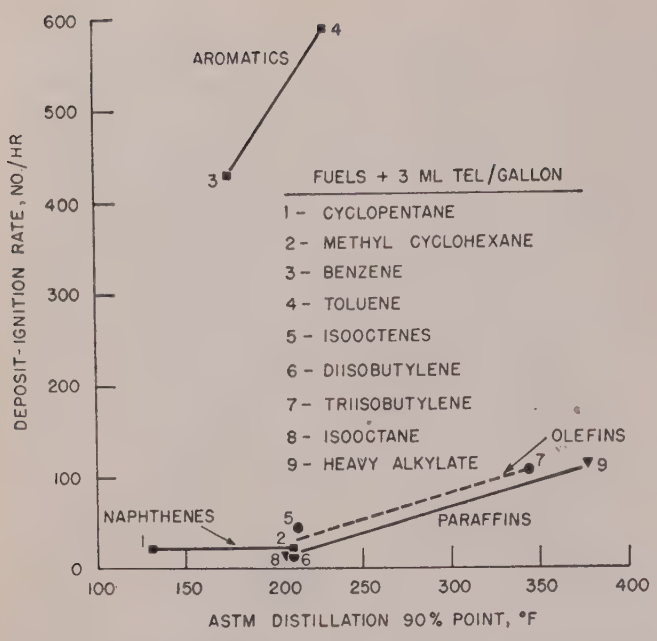

Fig. 9 – Effect of fuel hydrocarbon type and volatility on deposit ignition

varying the proportions of isooctane and normal heptane. Fig. 8 presents the deposit-ignition rates obtained with these fuels, and shows that the rate of deposit ignition was the same with the three fuels, all containing 3 ml tel, and ranging in Research octane number from 69 to isooctane + 3.5 ml tel. To substantiate these data, the effect of fuel octane number on deposit-ignition rate was determined after stabilized deposits, from the standpoint of requirement increase, had been accumulated. It was found that the deposit-ignition rate obtained with isooctane continuing 3 ml tel per gal did not change when the fuel was changed to a blend of 70% isooctane and 30% normal heptane plus 3 ml tel per gal.

The foregoing indicates that the deposit-ignition rate is independent of fuel octane number, although, as previously pointed out, the audible effects can be suppressed by higher-octane-number fuels. Since deposit ignition as measured by the counter is not appreciably influenced by fuel octane number, comparisons between fuels are greatly simplified. Consequently, this technique produces results which are more fundamental in nature and hence superior to those obtained by audible measurements, in which fuel octane number is a factor.

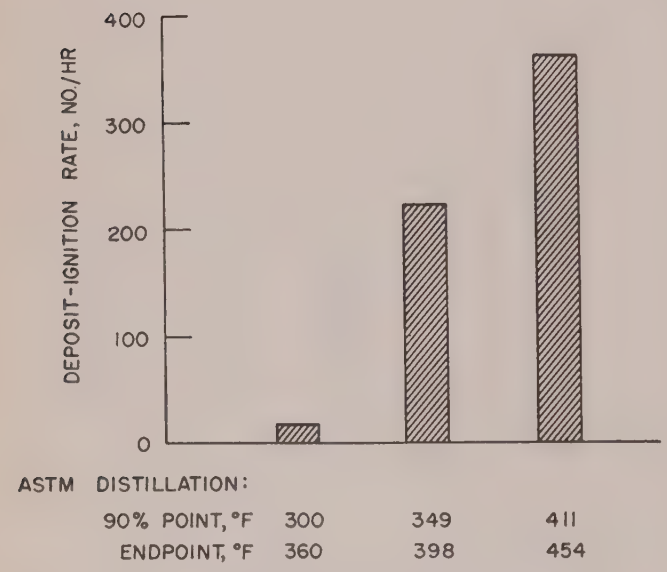

Fig. 10 – Relation of deposit ignition to gasoline end point. Fuel was catalytically cracked stock containing 3 ml tel per gal

Fuel Factors

It has been pointed out previously by Campbell, Withrow, and Bowditch[2, 3, 4] that a wide variety of chemical materials found in engines, including lead compounds and metallic oxides, can lower the ignition temperature of certain forms of carbon and thereby promote its burning. This catalyzed burning of carbonaceous material in deposits may generate sufficient heat to produce ignition of the fuel-air mixture. Ignition caused by this type of deposit would then appear to depend on the type and amount of carbon produced by the fuel and oil as well as the presence of catalyzing materials. These latter materials may originate from compounds present in the fuel, oil, air, or from wear of engine parts.

The effect of tetraethyl lead as motor antiknock compound (hereafter tel) was investigated in this mild, cycling-type laboratory test schedule in which efforts were made to minimize the presence of other catalytic materials. It was found that the effect of tel in promoting deposit ignition was highly dependent on base fuel type. For instance, the effect was very small in isooctane, quite large in toluene, and intermediate in commercial gasolines.

In examining the effects of fuel and oil composition, it was desirable to control the presence of the catalytic materials. Consequently, the studies reported below were conducted with fuels containing 3 ml tel, and nonadditive lubricating oils, unless otherwise noted. Furthermore, the amount of contaminants in the intake air was minimized by passing the air through a water-spray chamber.

The carbonaceous materials involved in this problem originate from both the fuel and the lubri-

[2] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-751 + (disc) 751-752: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.
[3] United States Patent 2,405,560 to J. M. Campbell.
[4] United States Patent 2,427,173 to L. L. Withrow.

cating oil. Consequently, fuel characteristics would be expected to have an important influence on the amount of deposit ignition. As a step toward determining the relative importance of several fuel characteristics, the effects of fuel volatility and hydrocarbon composition were examined.

Fuel Volatility and Composition – Several fuels were tested, which were representative of different individual hydrocarbon types, and which had different volatilities. Fig. 9 shows that the aromatics, benzene and toluene, produced by far the highest rates of deposit ignition. In comparison, the olefins, paraffins, and naphthenes all gave relatively low rates of deposit ignition. Furthermore, within a given hydrocarbon class, there was a general tendency for higher-boiling materials to produce more deposit ignition.

The significance of heavy fuel components, as measured by volatility, was further investigated by distilling a catalytically cracked naphtha to three different end points: 360, 398, and 454 F. The amount of deposit ignition obtained with these fuels was found to decrease rapidly as the heavier portions of the fuel were removed, as shown in Fig. 10. Since hydrocarbon-type proportions did not vary appreciably among these fuels of different end points, the effects observed appear to be due primarily to changes in volatility. It should be pointed out that, although this large effect of volatility has been investigated so far with only one gasoline stock, it is supported by the results with individual hydrocarbons. Since the effect is so marked, it represents an attractive field for further research with other gasoline stocks.

Comparison of Full-Boiling Gasolines – The large differences found among the hydrocarbon classes made it of interest to determine if differences among full-boiling fuels could be related to their hydrocarbon-type analyses. Three commercial gasoline blends and five stocks produced by different refinery processes were tested. These fuels all contained 3 ml tel per gal and had similar distillation curves with end points of approximately 400 F. The effects of these fuels on deposit ignition are compared in Fig. 11 with those of isooctane and toluene; these latter hydrocarbons produced the minimum and maximum rates found in the group of individual hydrocarbons. It may be observed that all of the gasolines were intermediate between the two hydrocarbons and that appreciable differences in deposit ignition existed among the gasolines. These differences, however, were not directly related to fuel composition as shown by the hydrocarbon-type analyses presented in the upper part of Fig. 11.

It may be concluded that the deposit-ignition tendencies of mixtures of complex hydrocarbons as encountered in full-boiling gasolines cannot be predicted accurately by measurement of their overall hydrocarbon-type proportions by the present standardized techniques. Additional study of mixtures including both simple and complex hydrocarbons would be desirable in order to provide a better understanding of the effects of fuel composition. Nevertheless, the foregoing studies have indicated that fuel volatility and composition are highly important factors in the deposit-ignition problem.

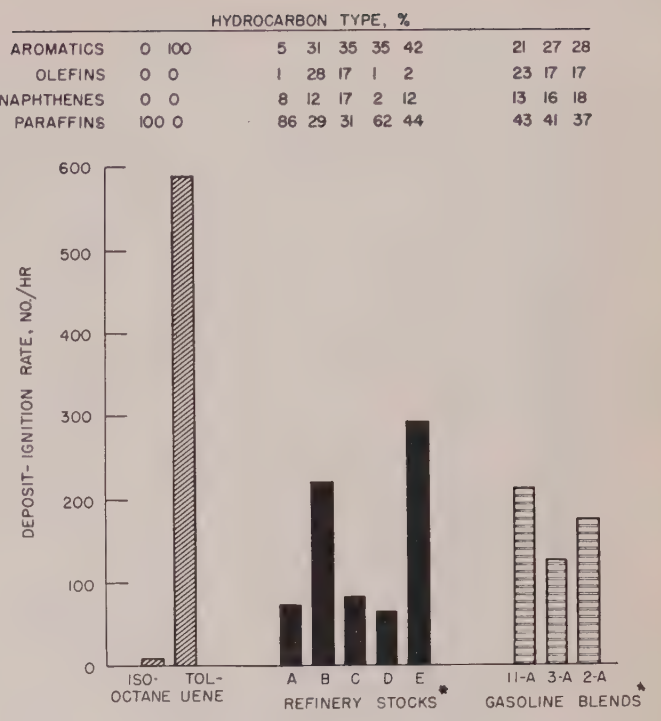

Fig. 11 – Effect of different fuels on deposit ignition. Fuels contain 3 ml tel per gal
***End points approximately 400 F**

Influence of Operating Conditions on Fuel Performance – It has been found that the differences

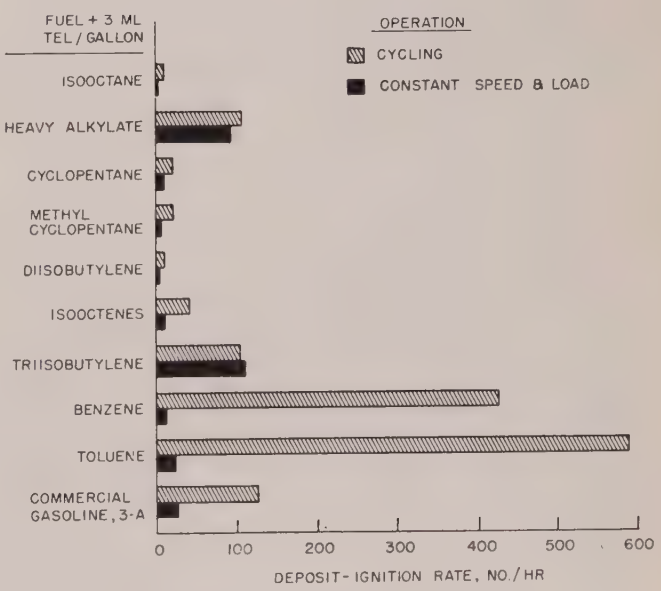

Fig. 12 – Relative effects of constant speed and load versus cycling operation on deposit ignition of various fuels

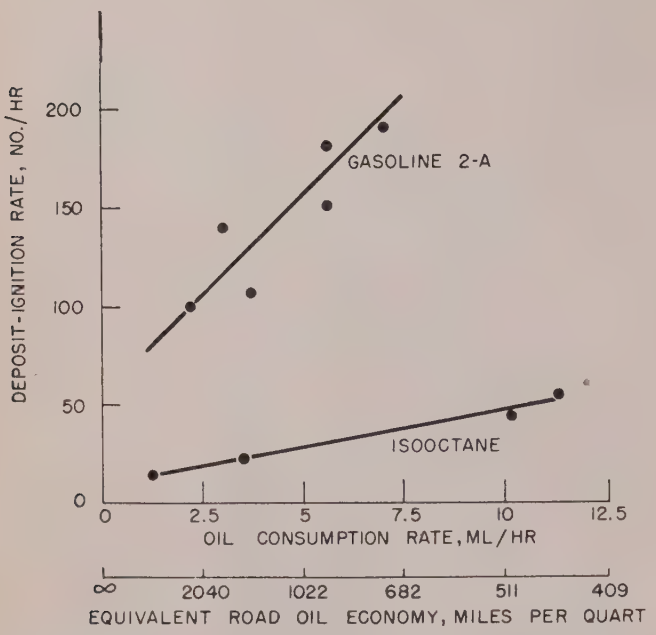

Fig. 13 – Effect of oil consumption rate on deposit ignition. Oil X, nonadditive. Fuels contain 3 ml tel per gal

among leaded fuels can be greatly influenced by engine operating conditions. A series of leaded fuels was tested both under the standard cycling test schedule and under operation at constant speed and load in which the idling portion of the cycle was omitted. This change tended to reduce carbon formation, since it eliminated the periods of rich mixture idling, acceleration, and deceleration which are conducive to incomplete combustion. These results are presented in Fig. 12. Although most of the fuels produced less deposit ignition when run under constant-speed, constant-load conditions, as would be expected, the greatest reduction was observed with the aromatic hydrocarbons, benzene and toluene, which are known to produce carbon readily.

From the foregoing, it may be concluded that cycling-type test conditions are advantageous for the study of deposit ignition. This type of test, by providing more opportunity for carbon formation, magnifies differences among fuels, and is believed to be more representative of normal passenger-car operation.

Table 2 – Analyses of Oils Y and X

	Oil Y	Oil X				
	Whole Oil	Whole Oil	Cut A	Cut B	Cut C	Cut D (Bottoms)
Volume Per Cent of Whole Oil	100	100	25.0	12.5	25.0	35.5
Distillation Range, F[a]	266–356+	316–419+	316–344	344–358	358–419	419+
Gravity, deg API	20.0	29.5	30.4	30.5	29.7	27.9
Viscosity:						
SUS at 100 F	475.0	307.0	108.5	138.4	204.6	620.4
SUS at 210 F	52.2	53.4	41.1	44.4	48.1	66.4
Sulfur, weight %	0.28	0.28	0.26	0.22	0.24	0.33
Phosphorus, weight %	0.025	0	0	0	0	0
Zinc, weight %	0.014	0	0	0	0	0

[a] At 0.080 to 0.140 mm Hg absolute pressure.

Oil Factors

Lubricating oil, as well as the fuel, contributes to the formation of combustion-chamber deposits. Most of the published information on this effect of oils has been concerned primarily with the weight of deposit formed. More recent work has pointed out that oil characteristics also affect the increase in ordinary knock caused by deposits.[1, 5] It might be expected, then, that deposit ignition would also be influenced by the lubricating oil. In the work discussed below, the effects of different base oils and of oil consumption rate were investigated.

Effect of Oil Consumption Rate – Fig. 13 presents the effect of oil consumption rate on deposit ignition with oil X. Data were obtained with two fuels containing 3 ml tel per gal, isooctane and a gasoline blend. Oil consumption rate was varied in these tests by changing piston-ring combinations. These data show that the rate of deposit ignition progressively increased with increasing oil consumption rate, and indicate that the lubricating oil contributes to the problem. Furthermore, the degree to which higher oil consumption rates increase deposit ignition is influenced by the fuel used, since the effect was much less with isooctane than with gasoline blend 2-A. These results also emphasize the importance of maintaining close control of oil consumption in test work.

Effect of Different Base Oils – In order to determine if a difference existed among base oils with regard to deposit ignition, two SAE 20 oils having widely different characteristics were compared. Fig. 14 shows that oil Y consistently produced less deposit ignition than oil X, with each of four different leaded fuels, which were isooctane, toluene, and two commercial gasolines. It appears then that differences among oils may be observed regardless of the fuel used. These data emphasize the importance of the lubricating oil in this problem and show that when an oil such as oil Y is used in conjunction with a favorable fuel, such as leaded isooctane, deposit ignition can be essentially eliminated.

Among the differences between these two oils are volatility, as shown in Table 2, and hydrocarbon composition. In an effort to study the effect

[5] "Combustion-Chamber Deposits as Related to Carbon-Forming Properties of Motor Oils," by J. D. Bartleson and E. C. Hughes. Presented at meeting of American Chemical Society, September, 1952.

of oil volatility alone, oil X was vacuum distilled into four fractions, and three of these fractions were tested with leaded isooctane for comparison with the whole oil. The results are presented in Fig. 15. Because of variations in oil consumption which were encountered in these tests, the deposit-ignition data are expressed as a percentage of the corresponding value obtained with the whole oil at the different oil consumption rates. Deposit ignition was found to become progressively less as the higher-boiling oil components were removed. This variation in deposit ignition is believed to be due primarily to the changes in oil volatility, since the hydrocarbon-type proportions of the fractions, as measured by ring analysis, were quite uniform throughout the distillation range.

Further confirmation of this volatility effect was observed when a high-boiling fraction amounting to 36% of whole oil X was removed by distillation. Removal of this fraction reduced the rate of deposit ignition to 75% of that obtained with the whole oil. This comparison was made using a commercial gasoline blend, 2-A, containing 3 ml tel per gal.

From the foregoing oil studies, it may be concluded that both the characteristics of the lubricating oil and the amount which enters the combustion chamber have a marked influence on deposit ignition. One factor which contributes to the differences among oils is volatility; lower-boiling oils reduce deposit ignition. The importance of other oil characteristics such as hydrocarbon composition and commercial additives has yet to be determined.

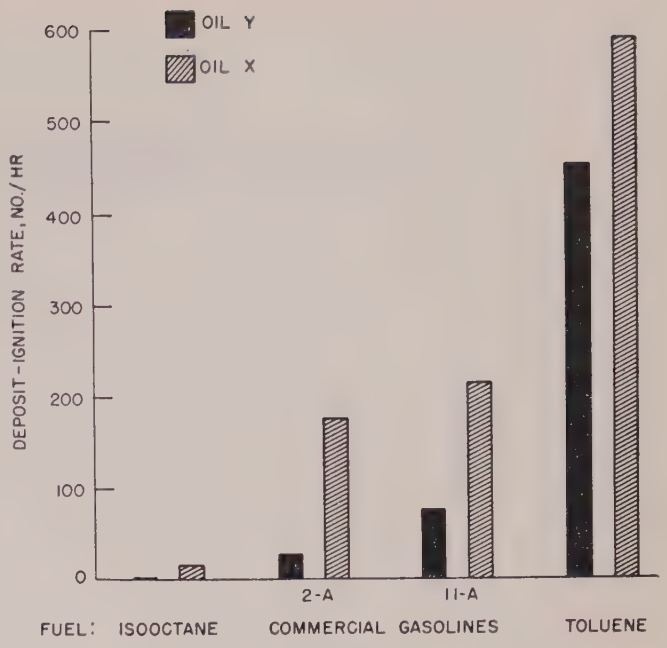

Fig. 14 – Comparison of deposit-ignition rates of two oils tested with four fuels. Fuels contain 3 ml tel per gal

Conclusions

A useful laboratory engine method has been developed for measuring deposit ignition (erratic ignition of the fuel-air mixture by combustion-chamber deposits). The method employs an ionization gap in the combustion chamber and electronic instrumentation which detects and records uncontrolled, deposit-ignited combustion. The method is superior to those previously used because it provides a continuous recording and is not dependent on audible effects, which do not always follow deposit ignition.

Laboratory investigations have been made to determine the effects of several factors, such as:

Fuel – Fuel composition and volatility affect the amount of deposit ignition. Reduction in fuel end point reduces deposit ignition. Among individual leaded hydrocarbons, aromatics produce by far the most deposit ignition. However, the differences among full-boiling gasoline stocks of similar volatility cannot be predicted from their overall hydrocarbon-type proportions.

Oil – Both the characteristics of the lubricating oil and the amount which enters the combustion chamber have a marked influence on deposit ignition. One factor which contributes to differences among oils is volatility; lower-boiling oils reduce deposit ignition. The amount of deposit ignition is directly related to the amount of oil entering the combustion chamber.

Operating Conditions – Engine operating conditions conducive to incomplete combustion and carbon formation tend to increase deposit ignition and magnify differences among fuels.

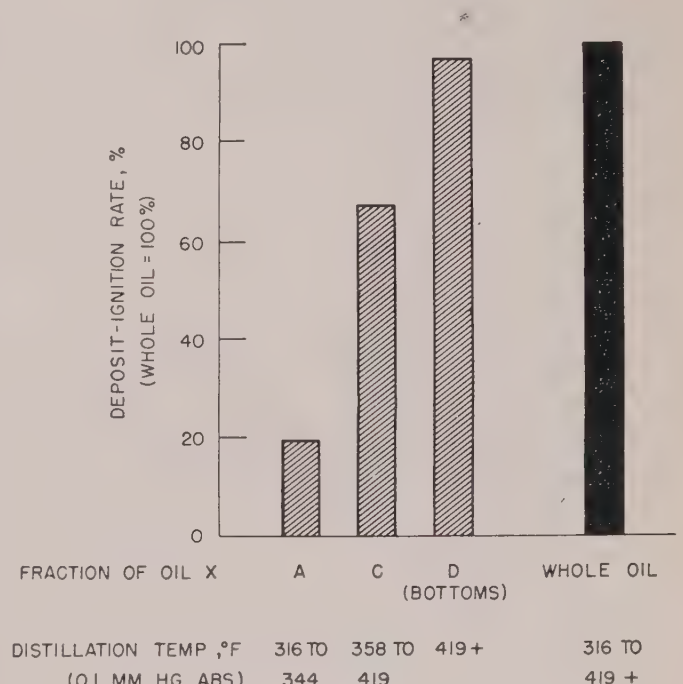

Fig. 15 – Effect of oil volatility on deposit ignition. Fuel is isooctane plus 3 ml tel per gal

Occurrence of Preignition in

THERE has been a lot of discussion about the prevalence of preignition in the newer high compression ratio engines, but very little factual data have been presented to indicate the seriousness or the extent of the problem. The purpose of this paper is to show that preignition is a problem in many cars that are now operating on the road.

Twenty-one 1952 cars of nine different makes and nine 1953 cars of six different makes have been evaluated to try to get some information on preignition and its relationship to knock. The bulk of this work was done with 1952 cars because of the unavailability of 1953 models at the time this program was started. Also, the 1952 cars were the most modern engines available with an essentially stabilized combustion-chamber-deposit condition. In cases where there were significant mechanical changes to the engines in the 1953 models, these cars were also evaluated.

The 1952 cars used were Cadillac, Chrysler, Packard, Oldsmobile, Buick, De Soto, Pontiac, Chevrolet, and Ford 6. The 1953 cars were Cadillac, Lincoln, Oldsmobile, Buick, Dodge, and Chevrolet.

Statement of the Problem

It is recognized that knock and preignition are separate and different phenomena; that they can both occur at the same time or that they can be present independently. It is also known that knock can bring on preignition and, most certainly, preignition can cause knock. These things have all been observed and substantiated by various methods in the laboratory, but there is

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PREIGNITION is defined by Mr. Winch as the initiation of a flame front by some means other than the spark discharge at some point in the combustion chamber prior to the arrival of the normal flame at that point. He believes that wild ping is actually a combination of preignition and knock.

His study of both 1952 and 1953 cars indicates that preignition has become a very serious problem. Moreover, in some cases, the problem was found to be more serious in the 1953 car of a particular make than in the 1952 model.

In these studies preignition and wild ping were identified by a technique that makes use of the spark plug as an ionization gap.

The Author

R. F. WINCH (J '52) is a research engineer in the Automotive Laboratory of the Sun Oil Co. He graduated from Swarthmore College in 1945 with a B.S. degree in mechanical engineering. Following graduation, Mr. Winch served in the United States Navy.

Present-Day Cars in Normal Service

R. F. Winch, Sun Oil Co

This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1953.

very little known about how the two phenomena occur in cars operating in normal service.

Knock has been the primary consideration in the design of combustion chambers and in the improvement of fuel. The type of operation which produces knock, the remedies available for eliminating it, and the means by which knock manifests itself in operating vehicles are all well known.

While it can be safely said that knock will continue to be a problem and that octane numbers, both chemical and mechanical, will always be sought, it is becoming more obvious that the problem of preignition must also be considered in future development work. It is important to know not only whether preignition is occurring but also how it is occurring, and its relation to knock. For this reason, three combustion phenomena are to be discussed in this paper: knock, preignition, and what is believed to be a combination of the two, wild ping.

Many attempts have been made separately to name and define the many types of abnormal combustion that may occur in an engine. Where test techniques are sufficiently sensitive so that these various combustions can be observed, such definitions are needed. However, most techniques now available are only able to indicate that something abnormal has occurred regardless of whether it was pre or post ignition. Strict definitions, then, do not seem to apply to studies utilizing these techniques. A more general definition would seem to be needed; a definition which will include all abnormal combustions other than knock regardless of cause, location, or timing. With this in mind, the occurrence of flame fronts initiated by causes other than the spark discharge have been arbitrarily called preignition. The following definition is included to justify this use of the word. Preignition is the initiation of a flame front by some means other than the spark discharge at some point in the combustion chamber prior to the arrival of the normal flame at that point. Knock does not fit this definition since it does not conform to conventional flame front type of combustion.

Wild ping is believed to be a combination of preignition and knock which probably occurs as follows: A preignition cycle is initiated very early in the compression stroke. The rate of pressure and temperature rise due to the advanced flame initiation is so great that spontaneous ignition conditions are reached before very much of the charge is consumed. Because of the critical conditions, the remaining charge is almost instantaneously inflamed, causing a very sudden release of energy. Resulting pressure waves set up the vibrations which produce the characteristic wild ping noise. The similarity between this sudden energy release and the phenomena associated with knock has suggested this part of the wild ping cycle and knock are one and the same thing, differing only in degree or intensity. This fact has been very well substantiated by Withrow and Bowditch.[1]

Test Technique

The test techniques used for the work described in this paper have permitted the separate evaluation of knock, preignition, and wild ping. The data have brought forth a lot of interesting information,

[1] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-751 + (disc) 751-752: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.

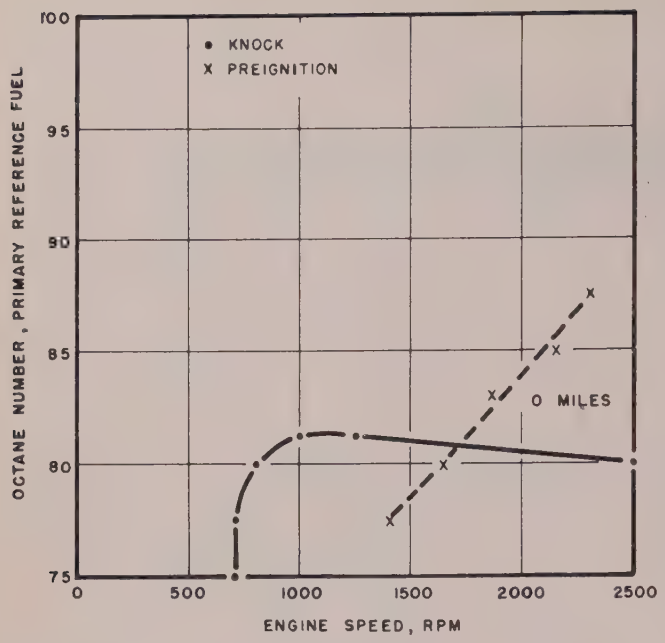

Fig. 1 – Speeds for knock and preignition during level-road acceleration – clean engine

but have also raised some questions, the answers to which are far from obvious.

The preignition and wild ping tendency of the cars were evaluated so that the relationship between these phenomena and knock could be shown. The most obvious way to do this would be to see how and when preignition occurred while determining the octane requirement of the car. The cars were operated in a manner very similar to the Borderline technique used for rating fuels and cars on the road. They were run through level-road, full-throttle accelerations from the lowest speed possible in high or fourth gear to 2500 rpm. During the accelerations, the speeds at which knock, preignition, and wild ping occurred were recorded. Knock was determined by ear and included any abnormal noise heard by the driver. No attempt was made to distinguish between knock and wild ping.

Preignition and wild ping were identified by the technique described in a previous paper.[2] Briefly, this procedure makes use of the spark plug as an ionization gap. Through oscilloscopic observation of the electrical activity within the ionization gap, it is possible to determine: (1) when an engine is preigniting and (2) when wild ping occurs. The technique is not dependent on any audible effect. It is capable of indicating preignition occurring both in the presence and absence of knock.

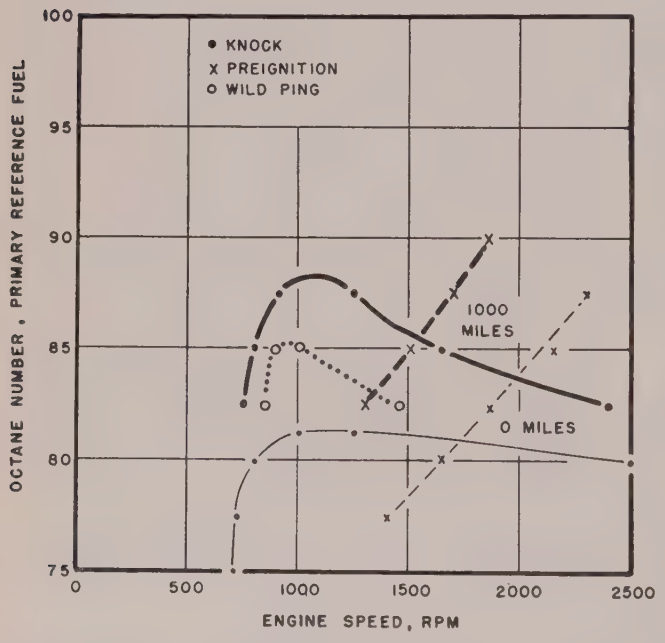

Fig. 2 – Speeds for knock, preignition, and wild ping during level-road acceleration – after 1000 miles of operation – data of Fig. 1 also included.

Fuels

The fuels used were selected so that some information could be obtained on the performance of these cars using two different types of fuel:

(1) Primary reference fuels were used for most of the work since preignition was to be studied in relation to knock.

(2) A group of premium commercial fuels was used to show the preignition performance of these cars on a more conventional fuel. Data obtained with one of these fuels will be discussed.

Significance of Data

In order to show that the procedure is capable of indicating differences in preignition tendency, it will be necessary first to describe in some detail the significance of the rating procedure used. This can best be done by observing the knock, preignition, and wild ping ratings obtained in one particular engine as it was run from a clean combustion-chamber condition to a heavy deposit accumulation.

Fig. 1 shows the engine speeds at which knock occurred with a series of primary reference fuels in a clean engine. Preignition also was observed while the octane requirement ratings were being obtained. The second curve shows when the preignition occurred on each fuel.

As an example, during an acceleration on 80-octane reference fuel, knock occurred at 800 rpm. As speed was increased, the knock continued. At 1650 rpm preignition was observed on the oscilloscope. Both knock and preignition were present until at 2500 rpm knock died out, but the pre-

[2] See SAE Transactions, Vol. 61, 1953, pp. 453-461 + (disc) 461-468: "A Method of Identifying Preignition," by R. F. Winch and F. M. Mayes.

ignition continued. 82.5-octane fuel was run next. There was no knock with this fuel. However, preignition occurred at 1850 and higher rpm. There was no audible effect from this preignition, which means that it would have been unnoticed by the driver.

Four things become apparent from these curves:

1. Preignition can occur in the absence of combustion-chamber deposits.

2. Preignition can occur in the absence of any audible effect. It can occur on fuels which are of a higher octane value than the requirement of the engine.

3. Preignition is mildly affected by primary reference fuel octane number in a clean engine. The example above showed preignition occurred at 1650 rpm on 80-octane reference fuel and at 1850 rpm on 82.5-octane reference fuel. On 87.5-octane fuel the speed for preignition had moved up to 2300 rpm. The reason for the effect of octane number on preignition tendency shown here is not clearly understood. Single-cylinder engine evaluations at our laboratory have indicated octane number to have no effect on inaudible preignition.

4. As engine speed increases, the tendency for the engines to preignite is greater.

As might be expected, there was no wild ping present at this clean deposit condition.

The car was then put back into normal operation and as combustion-chamber deposits accumulated, the octane requirement of the engine increased in a normal manner. After 1000 miles of operation the car was rated again. Fig. 2 shows the manner in which the three phenomena occurred with the combustion-chamber deposits present. In this figure, the curves from the preceding figure have been included for comparison. Knock has now become more severe. 88-octane-number reference fuel is now necessary for knock-free operation. The preignition tendency has also changed. Instead of preignition starting at 2300 rpm on 87.5-octane fuel as before, it occurred at 1700 rpm. The deposits have reduced the speed necessary for preignition by 600 rpm.

Wild ping also occurred at this time. It was identified by the particular type of pattern seen on the oscilloscope while observing for preignition. The driver made no attempt to note the wild ping. Everything that could be heard was called knock, just what the average driver would call it. It is apparent from an examination of the knock and wild ping curves that, although they are at a different level, they are similar in shape. This similarity suggests that wild ping is affected by octane number in a manner similar to knock. If an engine is knocking, a higher-octane fuel will reduce the intensity of the knock, and if further increased, will eliminate the knock entirely. Wild ping responds to increased octane number in the same way.

The similarity in the response of wild ping and knock to octane number is very interesting and actually quite logical. The noise associated with wild ping has been attributed to severe detonation or knock of a large volume of unburned charge. Since knock is involved in wild ping, it is not unreasonable to expect that wild ping would respond to changes in octane number in a manner similar to knock. If octane numbers will sup-

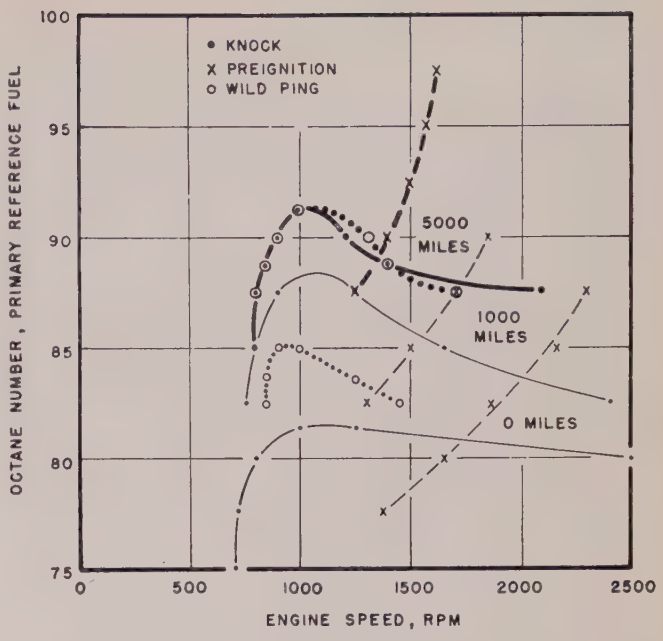

Fig. 3 – Speeds for knock, preignition, and wild ping during level-road acceleration – after 5000 miles of operation – data of Figs. 1 and 2 also included

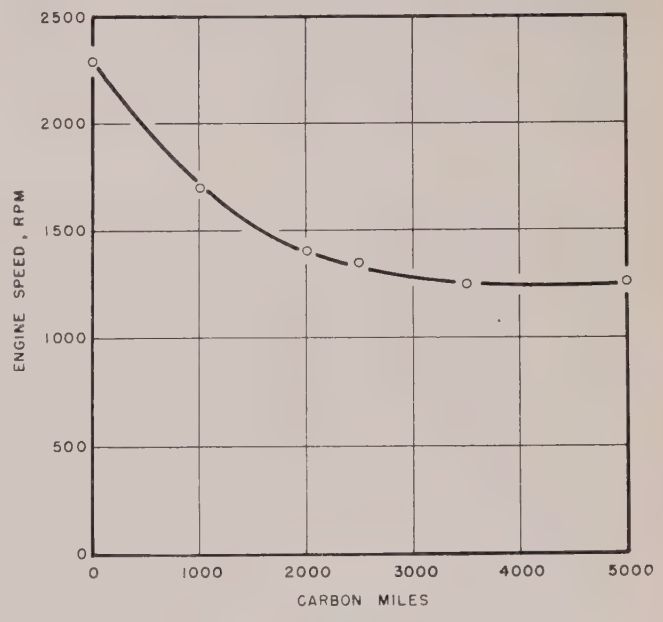

Fig. 4 – Preignition resistance with 87.5-octane primary reference fuel

press knock, they should also suppress the noise-producing reaction involved in wild ping.

At this point the question could be raised: If wild ping is a condition brought about by preignition, how can a separate preignition rating be obtained at the same time wild ping is occurring in the engine, as shown with 82.5 octane fuel in Fig. 2?

First, wild ping can be occurring in entirely different cylinders from those that are preigniting. In such a case it would be perfectly possible to get two distinct ratings at the same time, wild ping occurring in one cylinder while inaudible preignition was occurring in some of the others. Also, wild ping is very sporadic; once it has started it does not occur consistently nor always at the same speed. During the cycles when wild ping does not occur and preignition is present it can be separately rated as such.

To show how these ratings appear at one more deposit condition, the engine was rated again after the combustion-chamber deposits had been accumulated for 5000 miles. At this condition the three phenomena occurred as shown in Fig. 3. Here again, the curves from the previous two figures have been included for comparison.

The preignition tendency of the engine has changed markedly. Preignition on 87.5-octane-number fuel, for instance, which occurred at 2300 rpm when the engine was clean, now occurred at 1250 rpm. This indicates that combustion-chamber deposits have materially reduced the preignition resistance of this engine. These curves also demonstrate that the effect of octane number on preignition tendency tends to decrease when deposits are present in the engine. For example, an increase in octane number from 87.5 to 97.5 – 10 numbers – reduced the speed at which preignition occurred by only 400 rpm at the 5000-mile deposit condition. When the engine was clean, a 10-number increase raised this speed by 900 rpm. You will note that the knock rating and the wild ping rating are essentially the same curve and that 91-octane fuel is now necessary to prevent knock and/or wild ping. It was obvious from observing the indications on the oscilloscope and listening to the noise coming from the engine that, at this condition, most of the noise rated by the driver as knock was actually wild ping. Since wild ping is originated by preignition, the octane requirement of the car was being established by preignition and not knock.

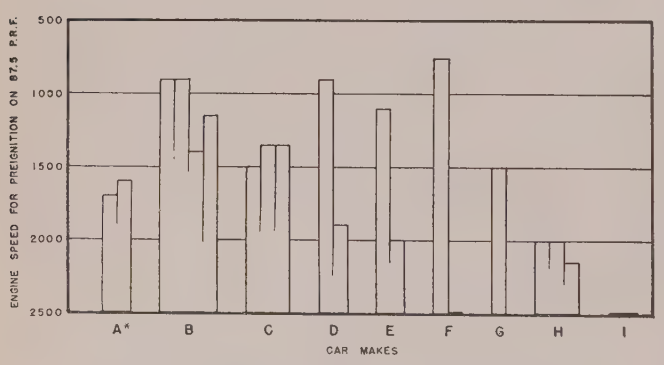

Fig. 5 – Preignition tendency of some 1952 cars
* Transmission limited lowest speed possible for these cars

The data used in this discussion demonstrate the changes that occur in a particular engine with the accumulation of combustion-chamber deposits. It should be emphasized that these curves are not necessarily of exactly the same shape nor do they assume the same relative position for every engine. Combustion-chamber configuration and fuel and lubricant properties, as reflected in combustion-chamber deposits, seem to exercise a significant influence on the shape and position of the curves. The relationships shown here are, nevertheless, generally representative of the manner in which these phenomena occurred in the cars evaluated during this program.

The next problem is, how can the preignition tendency of one engine be compared to that of another. A method is already available for comparative expression of knocking tendency by means of the octane scale. Fig. 3 indicates this same system can be used to express the wild ping tendency of an engine since wild ping seems to be another form of knock and responds to octane-number changes in much the same manner as knock. However, such a system of expressing preignition tendency has never been standardized. To give an engine a preignition rating which could be related to other engine conditions or even to other engines, we have arbitrarily picked the engine speed at which preignition occurs on 87.5-octane primary reference fuel. This method of expression was selected not because it was thought to be the best, but because it was felt to be simple, easily understood, and that it adequately described the relative performance of the various engines as determined by this procedure.

Fig. 4 shows the curve obtained with data taken from Fig. 3. This curve expresses the increased preignition tendency of an engine as deposits are accumulated. The figure shows that the 87.5-octane fuel rating dropped from 2300 rpm at the clean condition down to 1250 rpm after 5000 carbon miles. Since high engine speeds contribute to preignition tendency this reduction in speed for preignition indicates the engine is in a more serious preignition condition with deposits present.

Results

In studying the 30 borrowed cars, which are really the subject of this discussion, ratings were obtained in the "as received" condition. The mileages varied between 200 to 27,000 miles so that combustion-chamber deposits were not consistent.

The cars had been run on various fuels and lubricants and in widely varying kinds of service.

The data indicate that preignition is indeed a significant and serious problem. It was present in varying degrees in almost all of the 30 cars tested. Fig. 5 shows the speeds at which preignition occurred on 87.5-octane reference fuel. The scale on the ordinate is engine speed with zero at the top and 2500 rpm at the bottom. The height of the bar then indicates the range of the preignition.

The cars have been arranged by makes in the order of the severity of their preignition tendency on this fuel. Of the nine different makes of 1952 cars tested, preignition was detected in eight of them. However, there is a considerable spread in the resistance demonstrated by the different makes. Some of them, car B for instance, were preigniting almost as soon as the throttle was opened, while neither of the cars of make I showed any preignition difficulty at all on this fuel. In a few cases, there were wide differences in preignition tendency of two cars of the same make, such as makes D, E, and F. This is not too surprising since none of the cars were in the same combustion-chamber condition; they had not been driven in the same type of service, nor with the same fuels and lubricants.

Because of the transmission characteristics of car A, it was not possible to get a rating of the engine at speeds below 1700 rpm. As soon as the throttle was opened fully, engine speed went right to 1700 rpm. Preignition was present at this speed and was severe enough to indicate it would have been present at much lower speeds. For this reason it is placed as the most severe of the group.

The 1953 cars tested were, in some cases, improved over the 1952 models, while others were noticeably worse. The requirements of these cars are shown in Fig. 6. An actual comparison of the level of the 1953 model of car A with the 1952 model cannot be made because of the transmission, but, from observation, it is obviously in more trouble than is the 1952 model. Car H, a mild car in 1952, is apparently joining the ranks of the problem cars.

The knocking and wild ping tendencies of the 1952 cars are shown in Fig. 7. Here the cars are arranged in the same order as they were in Fig. 5. The cross-hatched bars indicate the octane number of the fuel necessary to prevent detonation, while the solid bars represent the octane number of the fuel necessary to prevent wild ping. It was found that cars with the highest preignition tendency in most cases were also highest in octane requirement. Make C is apparently an exception. With one car of this make there was no wild ping present at all. With some of the cars of makes A, B, and E, the wild ping ratings are as high as the knock ratings. As previously mentioned, this suggests that preignition in the form of wild ping, and not just detonation, is dictating the octane level of the

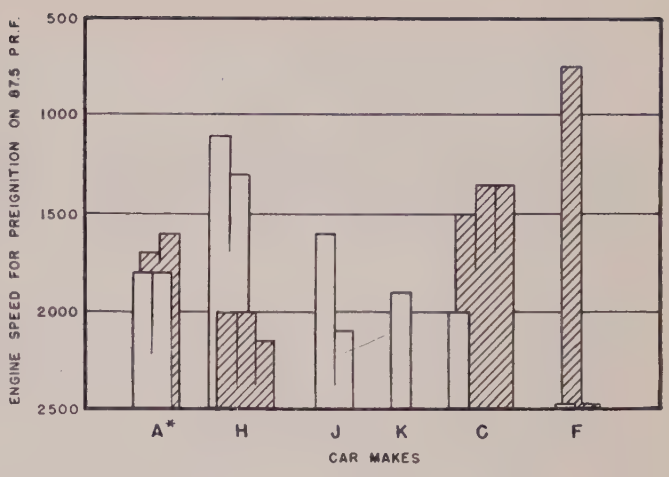

Fig. 6 – Preignition tendency of some 1953 cars – shaded bars are 1952 model data from Fig. 5

* Transmission limited lowest speed possible for these cars

fuel necessary to satisfy these particular engines.

There was a very large spread in the ratings obtained with the 1953 cars. As shown in Fig. 8, the rating of car A was exceptionally high and, with one car, was definitely limited by wild ping. Car H was shown to have a higher preignition tendency in 1953 than it did in 1952. The octane requirement is also a lot higher in the new model. Both cars of make H were in considerable wild ping difficulty. The other 1953 cars were relatively mild, makes K, C, and F showing no wild ping tendency at all.

The same 30 cars were also evaluated using a typical commercial premium fuel. The results of this work are summarized in Fig. 9, where the number of cars knocking, wild pinging, and preigniting on a typical commercial fuel is shown as a percentage of the total cars rated. Nearly all

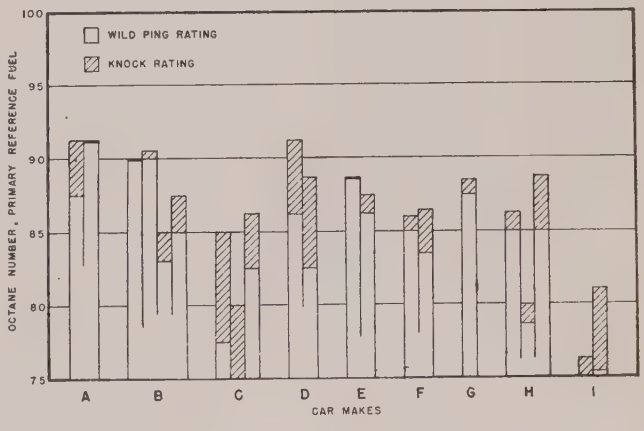

Fig. 7 – Knock and wild ping requirements of some 1952 cars

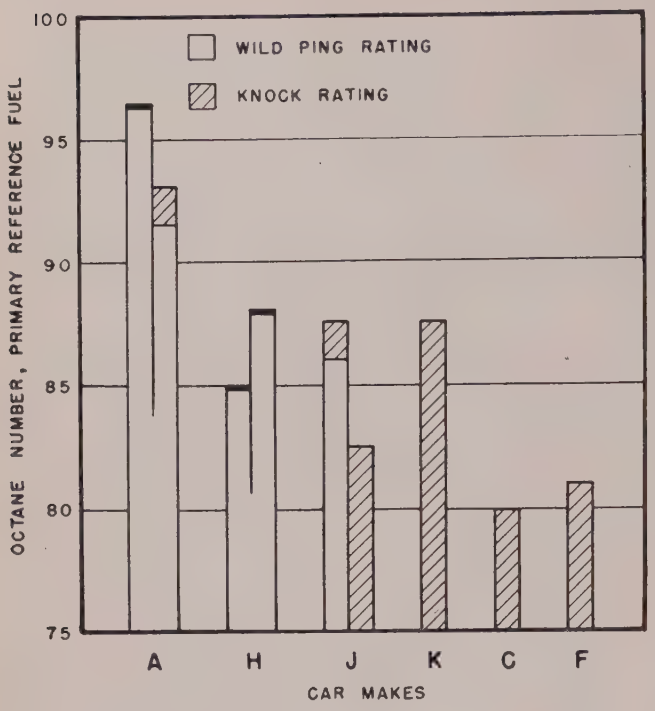

Fig. 8 – Knock and wild ping requirements of some 1953 cars

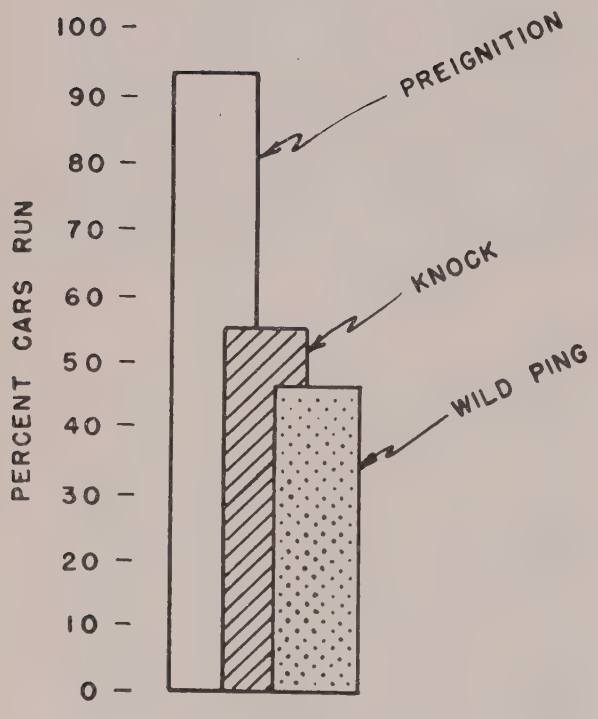

Fig. 9 – Premium commercial fuel performance

of the cars had some preignition occurring with this fuel, and over 50% were knocking. Wild ping was occurring with 45% of the cars run on this fuel. For this particular 45%, preignition is a very real problem, a problem that is now reaching the ears of the driver in the form of wild ping. From the results described herein, the old solution for preventing audible knock – octane numbers – is known to be rather ineffective in solving the basic preignition problem. Octane numbers will help suppress the noise of wild ping but are relatively ineffective in controlling the real trouble – preignition. Perhaps there is an easier, more economical, solution than the octane-number approach. If preignition could be eliminated or even reduced, it is conceivable that the present octane-number problem might be relieved or that future engine performance gains might be made without the customary octane requirement penalty.

There are two possible fields of investigation that might bring such relief:

1. The petroleum industry must look again at its fuels and its lubricants. Some basic studies are called for to find out what fuel property or properties determine preignition resistance of a fuel and how to increase this resistance. The extensive efforts being concentrated on reducing the effect of combustion-chamber deposits on octane requirement should also be concerned with the reduction of preignition. As was shown earlier in this paper, and as has been demonstrated by many others, combustion-chamber deposits have a very great influence on preignition. There is considerable justification for continued searches for means of reducing deposits.

2. Engine designers must become increasingly aware of preignition, both the audible and inaudible variety, as a problem and should consider it with as much respect and ingenuity as the detonation problem has received. As an example of the type of correction necessary in engine design, these two cases are cited. In one of the makes of cars evaluated, cylinders 3 and 7 were the preigniting cylinders in nearly all of the cars studied. If the difficulty in these two cylinders were corrected, the ratings would have been considerably lower. In another make, no two of the eight pistons had the same dome surface smoothness. The piston with the most pronounced surface-irregularities was consistently in the most serious preignition difficulty.

This information has been helpful to us in determining the seriousness of today's preignition difficulties and in helping to decide and justify our present and future studies of this problem. It has been presented because it is felt to be a timely subject, one in which many organizations are interested. It is hoped that it will serve the same purpose for others as it has for us, adding emphasis to the growing realization that preignition is a present-day problem.

Some Effects of Fuels and Lubricants on Autoignition in Cars on the Road

R. K. Williams and J. R. Landis, Research Laboratories Division, GMC

This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1953.

DURING recent months petroleum technologists and automotive engineers have expressed increased concern over problems of uncontrolled combustion associated with ignition of the fuel-air mixture by causes other than the spark. The physical characteristics of most forms of uncontrolled combustion have been recognized for nearly fifty years. Ricardo[1] in a presentation made in 1935 stated that as early as 1904 he and Hopkinson had observed differences in pressure development between knock and preignition. The former was accompanied by "metallic, ringing" sounds while the latter was characterized by "dull, thudding" noises. Similar differences in pressure development were recognized by early researchers in this country, as evidenced by the characteristic pressure-time cards presented by C. F. Kettering[2] before the SAE in 1919.

As a result of this pioneering work, the dependence of these phenomena on certain chemical characteristics of the fuel was soon discovered. Although considerable information is available on the effects of fuel and lubricant composition on the knock problem, little is known concerning the manner in which they influence other types of uncontrolled combustion. The material included in this investigation, obtained by means of road test procedures, deals with the effects of fuel and lubricant composition on types of uncontrolled combustion other than knock. These ignition phenomena are referred to as autoignition for lack of physical data which would permit them to be described more specifically.

Autoignition as it is used here first becomes apparent to the motorist through sounds of sharp, erratic knocking, or a rapid succession of dull thuds, or a combination of these sounds. On recognition of these symptoms it is often possible to turn off the electrical ignition and find that the engine will continue to run. The occurrence of autoignition was verified in this manner for the observations reported in this paper. While it is possible for autoignition to develop sufficiently to cause severe engine damage or a noticeable loss of power, the audible indications are generally detected before these more serious difficulties are encountered.

It has been found in current engines that autoignition of this type is usually observed only after sufficient operation to accumulate a layer of combustion-chamber deposit. While it is possible to induce autoignition in the absence of these deposits by operating for a time under severe knocking conditions, deposit-free operation is of little practical significance. Autoignition of this latter type is not included in this discussion of fuel and lubricant effects.

The type of operation during which autoignition is likely to occur differs for different engines depending upon the mixture, deposit, temperature, and pressure conditions prevailing in the engine cylinder. One factor is fairly clear, however. It is most prevalent during or immediately following

[1] See *Journal of Institute of Fuel*, Vol. 9, October, 1935, pp. 10-15: "Progress of Internal-Combustion Engine and Its Fuel," by H. R. Ricardo.

[2] See SAE Transactions, Vol. 14, 1919, Part I, pp. 201-219: "More Efficient Utilization of Fuel," by C. F. Kettering.

the transition from periods of light load operation to conditions requiring an increase in power output. The fact that engine deposits accumulated under light load tend to cause inflammation of the fuel-air mixture as they are removed from the combustion-chamber surfaces under heavy load conditions has been demonstrated by Withrow and Bowditch[3] by means of high-speed flame photographs. The conditions under which the erratic knocking or dull thudding is most prevalent in cars in normal road service suggest that this autoignition is caused by the phenomena described by these authors.

Another factor, and the one constituting the basis for this discussion, is the observation that the conditions under which autoignition can be detected seem to be influenced by the type of fuel and lubricant used during the deposit accumulation period.

Test Procedure

Two essential features of any program for evaluating fuel and lubricant effects on autoignition tendency are (1) a reproducible means of accumulating combustion-chamber deposits and (2) a reproducible means of measuring autoignition tendency.

Accumulation of Deposits – It has been our experience that autoignition tendency depends on the deposit accumulation schedule. The schedule used for these tests was one devised several years ago to measure the effects of deposits on knocking tendency. In this schedule the maximum operating speed is limited to 50 mph over a prescribed test course consisting of 15 miles of level road and 11 miles of hill operation. Car stops, transmission shift speeds, and operating speeds are rigidly specified for the various portions of the course. The deposits accumulated are believed to be representative of moderate driving conditions such as might be encountered during operation in cities and suburban areas.

Measurement of Autoignition Tendency – It has been found that autoignition of the type under consideration can be eliminated by increasing fuel antiknock quality. For this reason autoignition tendency was measured in terms of octane numbers. It was determined by testing the vehicle in a prescribed manner using reference fuels of progressively higher antiknock quality until two successive fuels were found, one giving autoignition at some point during the test sequence and the other giving autoignition-free operation throughout the entire test. Since conditions most likely to result in autoignition differ for different engine-transmission combinations, a variety of engine speeds and throttle openings were included in the test sequence. It has been found important in developing consistent results that the test steps be conducted in a definite order. The following procedure was used for these tests:

(a) Using a given reference fuel, a series of four accelerations is conducted on a level road. These accelerations are made at the maximum throttle opening in highest gear starting at 20 and terminating at 50 mph. The car is permitted to decelerate without braking between successive accelerations. Any evidence of autoignition verified by continuation of the engine to fire on cutting ignition constitutes a failure, and this procedure is then repeated using a fuel of slightly higher antiknock quality. If no autoignition is observed, the operations in (b) are conducted.

(b) Following the last acceleration in (a) the throttle is closed and the brakes are applied to reduce the car speed as rapidly as possible to 20 mph. Three accelerations similar to those in (a) are made using this braking procedure between each acceleration. If autoignition is detected, the operations are repeated using the reference fuel of next higher antiknock quality.

(c) Using the fuel giving autoignition-free operation in (b), the car is accelerated in highest gear at maximum throttle opening from 20 to 35 mph up a 7% grade. The throttle is held at the maximum opening and, on reaching 35 mph, the brakes are applied until the car speed is reduced to 20 mph. It is then permitted to accelerate at maximum throttle opening to 50 mph. If autoignition as described in (a) is encountered, the operations are repeated using the fuel of next higher antiknock quality until no autoignition is detected.

(d) Using the reference fuel giving autoignition-free operation in (c), the car is driven at 35 mph road load on an 11% grade. The throttle is then opened to the maximum position for 300 ft. Maintaining the open-throttle position, the car brakes are applied to reduce the car speed to 30 mph, from which point the car is permitted to travel 300 ft without braking. The speed is further reduced to 25 and 20 mph and observations of autoignition are repeated as indicated above.

(e) Using the reference fuel giving autoignition-free operation in (d), the car is then driven in highest gear at 20 mph on a 7% grade using sufficient throttle opening to maintain constant car speed for 600 ft. The car is then accelerated gradually in such a manner that 30 mph is reached in the next 600 ft. It is then accelerated at the maximum throttle opening to 50 mph. The purpose of this phase of the operation is to explore the range of low-speed operating conditions in which mixture composition and spark advance are changing from those used at part load to those required for maximum power. Autoignition observations are made as indicated above.

(f) Using the reference fuel giving autoignition-

[3] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-751 + (disc) 751-752: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.

STUDIES of cars on the road show that both fuel and lubricant characteristics affect the occurrence of autoignition.

For instance it is indicated that autoignition can be eliminated by increasing the fuel antiknock quality, although the Research octane number required may vary slightly depending on fuel composition. Limited evidence is also presented to indicate that aromatic constituents of gasoline are somewhat less effective in suppressing autoignition than are paraffin hydrocarbons.

In addition, it is shown that differences in autoignition tendency are observed when different fuels and lubricants are used during the deposit buildup period. In one series of tests a difference of seven octane numbers was observed between autoignition tendencies resulting from operation with different lubricants.

Further tests indicate that the addition of the amount of tricresyl phosphate theoretically required to convert all the lead in the fuel to lead phosphates lowers the Research octane number required to eliminate autoignition by about three octane numbers.

The Authors

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J. R. LANDIS (J '52) is leader of the multicylinder engine testing group of the Fuels and Lubricants Department of the Research Laboratories Division, General Motors Corp. He began work at the Research Laboratories in the field of road knock testing of automobiles after graduation. Mr. Landis attended both the University of Illinois and Lawrence Institute of Technology, receiving a B.S. degree in chemical engineering from the latter school.

free operation in (e) and using the brakes where necessary to maintain 35 mph for 300 ft on an 11% grade, the throttle is gradually opened and closed to explore part-throttle autoignition characteristics. This procedure is repeated at 30, 25, and 20 mph. Any signs of autoignition are recorded.

The autoignition tendency of the car for a given observation is described in terms of two fuels, the first being the fuel of lowest Research octane number giving autoignition-free operation, and the second being the fuel of highest octane number with which autoignition was detected at some point during the test.

Modification of certain portions of the test procedure are necessary to accommodate all car makes. However, as long as a uniform procedure is followed for a given design, reproducible results can be obtained. In some instances it may be desirable to eliminate certain features of the above procedure, depending on the type of information desired. This also has been found satisfactory, provided a standardized technique is followed; however, the autoignition level finally established will depend on the severity with regard to speed and load conditions of the particular test chosen.

Determination of Borderline Autoignition Tendency – For a given car and deposit accumulation conditions results of periodic measurements of autoignition tendency show certain variations. This is true even though deposits are replaced by additional driving between tests. This variation has been attributed to cyclic variations in deposit condition such as have been mentioned by Moller and Moir,[4] to effects of changes in atmospheric condition, to unknown factors, and to experimental error. Therefore, it has been found advisable to make a number of determinations of autoignition tendency and to replace the deposits removed during the process by additional driving between determinations. If the results of these observations are evaluated statistically, it is felt that the effects of the unknown and uncontrolled factors are minimized and that a reliable measure of autoignition tendency is obtained.

A typical set of observations conducted according to the procedures already described is shown in Fig. 1. These results were obtained in a 1952 L-head engine at 7.7/1 compression ratio. Deposits were accumulated using a commercial premium gasoline containing tel (tel as used herein refers to the tetraethyl lead contained in commercial Ethyl fluid) and a paraffinic-base, SAE 20W lubricating oil. The car was driven 500 miles between successive measurements of autoignition tendency. The upper boundary of each shaded area indicates the minimum Research octane number of the reference fuel giving autoignition-free operation for a particular measurement, while the lower boundary indicates the fuel of maximum antiknock quality with which autoignition was observed.

In Fig. 2 the reference fuel Research octane

[4] See SAE Transactions, Vol. 55, 1940, pp. 250-261: "Engine Deposits and the Effect of Some Fuel Additives," by J. A. Moller and H. L. Moir.

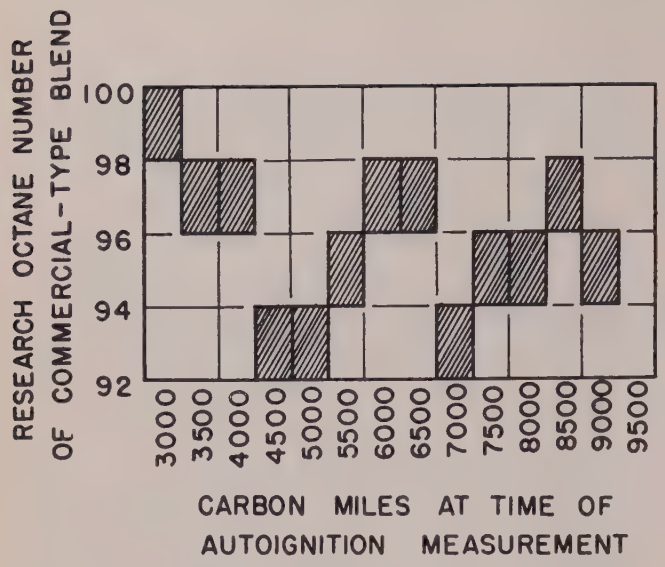

Fig. 1 – Chronological record of autoignition tendency. 1952 L-head engine; 7.7/1 compression ratio, leaded premium-grade tank fuel; paraffinic-base 20W oil

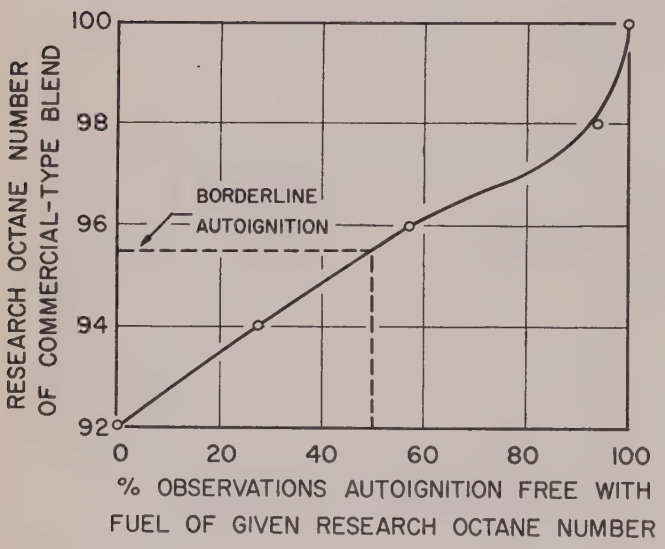

Fig. 2 – Statistical analysis of autoignition tendency. 1952 L-engine; 7.7/1 compression ratio; leaded premium-grade tank fuel; paraffinic-base 20W oil

Table 1 – Properties of Severity Reference Fuels

Fuel No.	Composition			Octane Number	
	% Iso-octane	% n-Heptane	% DIB	Research	Motor
11-S	50.5	11.6	37.9	100.0	88.3
10-S	42.3	16.9	40.8	97.5	86.6
9-S	34.5	21.9	43.6	95.0	84.7
8-S	27.4	26.5	46.1	92.5	82.9
7-S	25.4	29.7	44.9	90.0	81.4
6-S	26.0	32.2	41.8	87.5	80.1
5-S	26.6	34.5	38.9	85.0	78.7
4-S	27.1	36.7	36.2	82.5	77.2
3-S	29.8	38.0	32.2	80.0	75.7
2-S	40.7	36.4	22.9	77.5	74.4
1-S	49.7	35.0	15.3	75.0	72.9

number is plotted as a function of the percentage of measurements for which that fuel would operate without autoignition. Borderline autoignition tendency is determined from the octane number at the 50% satisfied point, which in this case was approximately 96. In determining borderline autoignition tendency in this manner, it is important that all observations be conducted after a reasonable combustion-chamber deposit level has been obtained. Tests have shown that 3000 miles of driving according to the schedule previously described provides a satisfactory starting point under most conditions.

Autoignition tendency is reduced as it is being measured due to the removal of deposits. Evidence that it is quickly restored during the course of normal deposit accumulation was shown in conjunction with tests conducted to evaluate a severe knocking treatment as a means of reducing autoignition tendency. Prior to the tests, a 1952 overhead-valve engine had been driven 13,000 miles using a paraffinic-base SAE 10W motor oil. Following measurement of autoignition tendency, a substantial portion of the deposits was removed by placing the transmission in neutral and accelerating the engine from 400 to 2400 rpm repeatedly until 1000-cc of normal heptane had been consumed. The deposit condition before and after this treatment is shown in Fig. 3. In Fig. 4 are shown the results of autoignition measurements before and for a time following this operation. While at first borderline autoignition tendency was reduced from 95 to approximately 86, the original level was restored within 350 deposit miles. Thus, these tests indicate that measurements of autoignition tendency can be made at 500-mile intervals without materially affecting the results once a reasonable level of deposit has been established.

Fuel Antiknock Quality and Composition

Once engine deposits resulting from operation with a particular combination of fuel and lubricant have developed to the point that autoignition can be detected, the fuel properties by means of which it can be eliminated are of interest. For example, one may wonder whether improvement of fuel antiknock quality by the addition of paraffin or aromatic components would be more effective in eliminating autoignition. This matter is of great importance where autoignition is limiting satisfactory operation of engines in view of the high cost of raising the antiknock quality of fuels from their present levels.

The relative significance of Research and Motor octane number in describing autoignition tendency of fuels was studied by conducting measurements in cars, stabilized with respect to autoignition tendency, using three different types of reference fuels. Identical tests were run in two different

overhead-valve, V-8 engines, one at 7.5/1 and one at 8.5/1 compression ratio. Deposits in both cars were accumulated using a commercial, premium-grade gasoline containing tel and an SAE 20W, mixed-base, solvent-refined, distillate oil treated to the MIL-0-2104 detergency level. A 70-mph limit cross-country schedule was used between measurements of autoignition tendency. Primary reference fuels, severity reference fuels, and a series of full-boiling-range gasolines were used for the measurements. The primary reference fuels were blends of isooctane and normal heptane graduated in 2.5-octane-number increments. The severity reference fuels were blends of isooctane, normal heptane, and diisobutylene. Their compositions and octane ratings are given in Table 1.

The full-boiling-range reference fuels were prepared by blending a high-octane-number commercial-type gasoline, a commercial premium-grade and a commercial regular-grade gasoline in such a manner as to provide fuels graduated in steps of approximately two octane numbers. Fuels above 92 Research octane number were obtained by blending the high-octane and premium gasolines, while fuels of lower octane quality were blends of the premium- and regular-grade fuels. Typical inspection data for the three base fuels are given in Table 2.

The results of these tests are given for each of the reference fuel series in Table 3. They are expressed in terms of the Research and Motor octane numbers for borderline autoignition as determined from plots similar to Fig. 2. The result for each fuel is based on 13 measurements of autoignition tendency.

It can be seen from the tabulation that the maximum spread in Research octane number for borderline autoignition is 2.2 octane numbers. This is evidence that octane number can be used as a measure of autoignition tendency. Since in both engines autoignition tended to occur at low speeds where the fuel road ratings correspond more closely to their Research than to their Motor ratings, it is not surprising to find that Research octane number provides the better measure of autoignition tendency. In view of the differences shown in Table 3, it appears that there may be a small effect of fuel composition, however.

In order to determine the effects of aromatic content on the autoignition tendency of a fuel, tests were run in which measurements were made in a 1952 overhead-valve V-8 engine at 7.5/1 compression ratio using primary and full-boiling-range reference fuels both with and without benzene additions. The primary and full-boiling-range reference fuels have been described above. In the blends of benzene and primary reference fuels 25% by volume of benzene was used in all members of the series, the balance of the blend consisting of the primary reference fuel required to give the desired Research octane number. Typical

Table 2 – Inspection Data for Full-Boiling-Range Blending Components

	High Octane	Premium	Regular
Specific Gravity at 60 F	0.73	0.74	0.74
Reid Vapor Pressure	8	8.0	9.0
Tel, ml/gal	2.76	2.0	1.5
Research Octane Number	100	92	86
Motor Octane Number	87	82	79
Distillation ASTM, F			
Initial	102	106	104
10% Point	136	134	133
50% Point	208	226	226
90% Point	318	336	320
End Point	368	391	383
Approximate Analysis, % by volume[a]			
Paraffins	40	63	73
Olefins	40	23	15
Aromatics	20	14	12

[a]Silica gel adsorption.

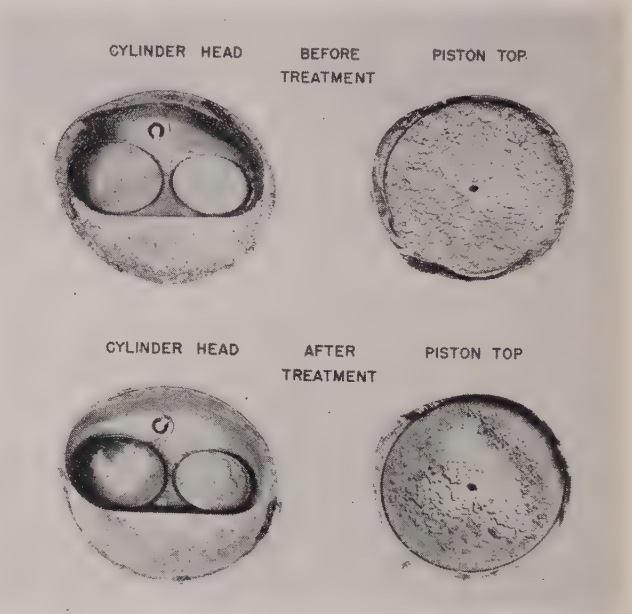

Fig. 3 – Combustion-chamber deposits before and after severe knocking treatment. 1952 overhead-valve engine; 7.5/1 compression ratio; leaded premium-grade tank fuel; paraffinic-base 10W oil

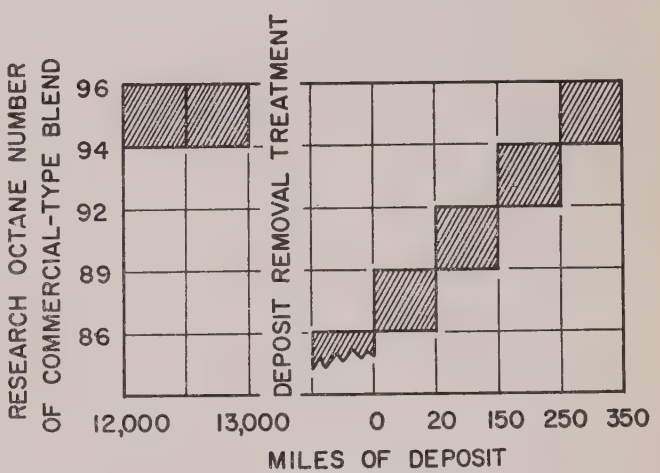

Fig. 4 – Chronological record of autoignition tendency before and after deposit removal treatment. 1952 overhead-valve engine; 7.5/1 compression ratio; leaded premium-grade tank fuel; paraffinic-base 10W oil. Fuel is autoignition-free at top of block; fuel autoignites at bottom of block

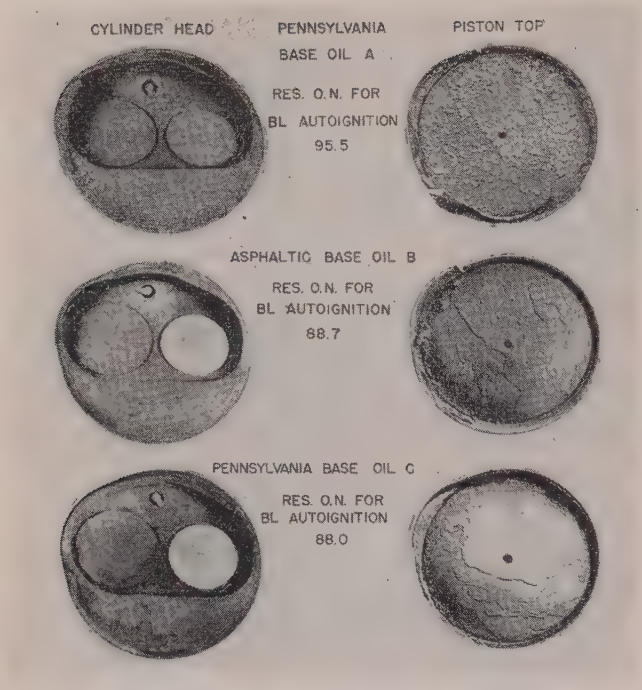
CYLINDER HEAD
PENNSYLVANIA BASE OIL A
RES. O.N. FOR BL AUTOIGNITION 95.5
PISTON TOP
ASPHALTIC BASE OIL B
RES. O.N. FOR BL AUTOIGNITION 88.7
PENNSYLVANIA BASE OIL C
RES. O.N. FOR BL AUTOIGNITION 88.0

Fig. 5 – Combustion-chamber deposits. 1952 overhead-valve engine; 7.5/1 compression ratio; leaded premium-grade tank fuel; 50-mph limit driving

Table 3 – Comparison of Different Reference Fuels in Measuring Autoignition Tendency

	Octane Number for Borderline Autoignition			
	7.5/1 Compression Ratio Engine		8.5/1 Compression Ratio Engine	
	Research	Motor	Research	Motor
Primary Reference Fuel	86.0	86.0	93.6	93.6
Severity Reference Fuel	84.6	78.3	94.0	84.1
Full-Boiling Range Reference Fuel	86.8	79.4	95.0	84.5
Spread	2.2	7.7	1.4	9.5

Table 4 – Properties of Blends of 25% Benzene and 75% Primary Reference Fuels

Octane Number		% by Volume in Blend		
Research	Motor	Benzene	Isooctane	n-Heptane
91.5	88	25	64	11
93.0	89	25	65	10
95.0	91	25	67	8
97.5	92	25	70	5

Table 5 – Properties of Blends of Benzene with Commercial Premium Gasoline

Octane Number		% by Volume in Blend	
Research	Motor	Benzene	Commercial Premium
92	82	0	100
93	83	10	90
94	83.5	20	80
98	86	40	60

blend properties are indicated in Table 4.

To provide a series of full-boiling-range fuels containing benzene, the antiknock quality of the premium fuel of Table 2 was increased by adding benzene to provide fuels of the desired octane steps. Typical compositions are given in Table 5.

The results of these tests, obtained using a paraffinic-base oil and commercial premium gasoline during the deposit accumulation, are given in Table 6. Autoignition tendency was measured using a different reference fuel series between each 500 miles of deposit accumulation until seven observations were obtained with each series. The seven measurements were combined in the manner indicated in Fig. 2 to give one value of borderline autoignition tendency.

A great deal more testing would be required in order to permit general conclusions to be drawn from these data. When considered in the light of the information presented in Table 3, however, certain trends are suggested. Considering the wide range in fuel composition involved, the spread in Research octane number of the fuels giving borderline autoignition in any one of these particular engines is not large. Thus, resistance of a fuel to autoignition is determined in part by its antiknock quality.

A second observation of interest is that in these particular tests fuels containing aromatic components seem to have slightly greater autoignition tendencies than aromatic-free blends. This trend is indicated by the following observations made from the data in Tables 3 and 6:

(a) In Table 3 the Research octane number of the full-boiling-range reference fuel required to suppress autoignition was greater than for either the primary or severity reference fuels, which contained no aromatics.

(b) Raising the antiknock quality of either primary reference fuels or the commercial premium-grade gasoline in Table 6 by adding benzene increased the Research octane number necessary to suppress autoignition. It should be pointed out that in no case were the differences large, however.

Fuel and Lubricant Composition

Lubricating Oil Effects – It is generally accepted that driving under light load conditions tends to create deposits which, as they are purged from the combustion chamber during periods of heavy load operation, increase autoignition tendency. Gibson, Hall, and Hirschler[5] have indicated the importance of the relationship of fuel and lubricant characteristics to deposit formation and

[5] See SAE Transactions, Vol. 61, 1953, pp. 361-376 + (disc) 376-377: "Combustion-Chamber Deposition and Knock," by H. J. Gibson, C. A. Hall, and D. A. Hirschler.

knocking tendency. Yet much remains to be learned about the influence of these variables upon the autoignition evidenced as the combustion-chamber deposits are swept from the engine. For this reason a number of tests were run in an attempt to determine the magnitude of differences in autoignition tendency resulting from operation with lubricating oils of widely different characteristics.

Three different cars of one 1952 make at 7.5/1 compression ratio were chosen for the tests. In order to add practical significance to the test data, the commercial premium-grade gasoline of Table 2 was used while deposits were being accumulated according to the 50-mph limit schedule. Borderline autoignition was measured in terms of full-boiling-range reference fuels using the procedure previously outlined. The physical properties of the lubricants along with results of borderline autoignition measurements are given in Table 7.

Because the presence of additives and V.I. improvers makes the volatility data questionable, particularly in the region of the end point where volatility effects are most likely to be significant, vacuum distillation data have been omitted from the tabulation.

Measured in terms of Research octane numbers the borderline autoignition tendency using oil B was almost seven numbers less than that when oil A was used. Since oil B was an asphaltic-base oil free of bright stock and oil A contained bright stock, this difference is consistent with the observations of Gibson, Hall, and Hirschler[5] and Bartleson and Hughes[6] on knocking tendency. The results with oil C, a narrow distillation cut paraffinic-base oil without bright stock, were equivalent to those obtained with oil B. This is further evidence that harmful autoignition effects are associated with the physical and chemical characteristics of the high-boiling and not the more volatile components of the lubricating oil.

Representative combustion-chamber deposits from operation with oils A, B, and C are shown in Fig. 5. It is evident that the deposits with oil A are heavier and of a different texture than those obtained with oils B and C, a factor which may be related to their autoignition tendencies.

Effect of Tetraethyl Lead – Gibson[7] has shown that the addition of tel to gasolines causes no significant difference in the increase in knocking tendency of cars in normal road service due to combustion-chamber deposits. However, it is possible that there may be a difference in autoignition tendency resulting from operation with leaded as compared to unleaded fuels. Because of the increased importance of autoignition in determining the antiknock quality of fuels required to satisfy the needs of higher compression ratio engines, this possibility was investigated.

Table 6 – Effect of Fuel Composition on Octane Number for Borderline Autoignition

	Approximate Composition,[a] % by Volume			Octane Number for Borderline Autoignition	
	Paraffins	Olefins	Aromatics	Research	Motor
Primary Reference Fuel	100	—	—	94.8	94.8
Primary Reference Fuel + Benzene	75	—	25	96.2	92.0
Full-Boiling-Range Reference Fuel	54	30	16	95.5	83.7
Full-Boiling-Range Reference Fuel + Benzene	44	16	40	96.5	85.2
			Spread	1.7	11.1

[a] Silica gel adsorption.

Table 7 – Effects of Lubricating Oil on Borderline Autoignition Tendency of 1952 Car Make at 7.5/1 Compression Ratio

Oil Base	A Paraffinic	B Asphaltic	C Paraffinic
Viscosity, SUS			
100 F	186	202	260.8
210 F	46	46.5	58.0
Viscosity Index	105	97	142
Additive			
Detergent	Yes	Yes	Yes
Inhibitor	Yes	Yes	Yes
V. I. Improver	No	Yes	Yes
Number of Tests (12 Observations per Test)	3	2	1
Average Research Octane Number for Borderline Autoignition	95.5	88.7	88.0
Range of Results	0	88.0 to 89.5	—

The tests used to determine the contribution of leaded fuel deposits to autoignition tendency differed slightly from other tests described in this paper in that consideration was given to the fuel requirement for suppressing knock as well as autoignition. This required a slight modification of the autoignition test procedure described previously so that the minimum fuel antiknock quality required to suppress either knock or autoignition, whichever was the limiting factor, could be determined. By this modified test the antiknock quality of the fuel for acceptable operation was established on the basis of knocking tendency if knock of objectionable intensity occurred using fuels of higher octane number than those required to eliminate autoignition.

The base fuel chosen for the tests was a commercial-type gasoline without tel. This fuel was used clear in one car and with the addition of 2.0 ml tel per gal in the other during the deposit buildup period. The two cars in which observations were made incorporated 9/1 compression ratio, experimental, V-8 engines of identical design. Combustion-chamber deposits were accumulated using the 50-mph limit deposit schedule. A mixed-base solvent-refined, distillate oil shown by previous experience to provide intermediate effects on knocking and autoignition tendencies was used

[6] "Combustion-Chamber Deposits as Related to the Carbon-Forming Properties of Motor Oils," by J. D. Bartleson and E. C. Hughes. Presented at a meeting of the American Chemical Society, Atlantic City, September, 1952.

[7] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 557-566 + (disc) 567-570: "Factors Affecting Octane-Number Requirement," by H. J. Gibson.

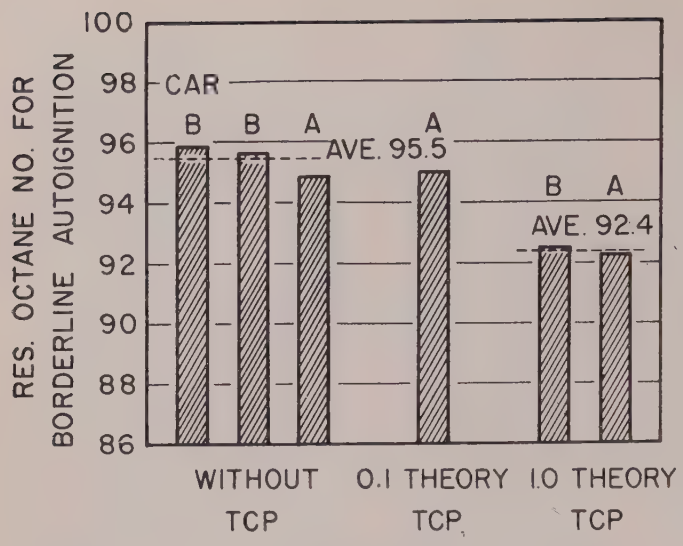

Fig. 6 – Effect of tricresyl phosphate on autoignition tendency

throughout the tests. In order to avoid contamination of the deposits from the unleaded fuel, observations in both cars were made using the severity reference fuel blends described in Table 1. These fuels did not contain lead.

The results of 15 determinations of the antiknock quality of the severity reference fuel required for satisfactory knock – and autoignition-free performance are given in Table 8.

The car operated on unleaded fuel was *knock-limited* and required a fuel of 92 Research octane number. At the expense of some power loss the fuel antiknock requirement of this car could be reduced by retarding ignition timing. In contrast, the car operated with the leaded fuel was *autoignition-limited* and required a fuel of 95.5 Research octane number for satisfactory operation. In this car retarding ignition timing would produce no reduction in fuel requirement.

In these tests deposits from the leaded as compared with the unleaded fuel materially affected the octane number of the fuel required for acceptable operation. The net effect of the addition of tel to the fuel during deposit buildup was an increase in the fuel antiknock requirement of 3.5 octane numbers. The addition of tel also resulted in a change in the limiting operating factor from knock, which could be eliminated by mechanical adjustment of ignition timing, to autoignition, which could be suppressed only by an increase in fuel antiknock quality.

Effect of Phosphorus Compounds – Experiments conducted at the General Motors Research Laboratories prior to World War II have shown that lead phosphates are relatively inactive as catalysts in the combustion of carbon. These early experiments led to the addition of trimethyl phosphate and tributyl phosphite to gasolines containing tel in order to explore in engines their effects on autoignition resulting from combustion-chamber deposits.[8] Although the beneficial effects of these additives on autoignition were quite marked, as were their effects on spark-plug resistance, the experiments were discontinued because of the formation of heavy exhaust-valve deposits.

Recent observations of the beneficial effects of tricresyl phosphate (tcp) as a spark-plug antifouling agent in aircraft[9] led to experiments to evaluate the effects of this particular phosphorus compound on autoiginition in automotive-type engines. The effects of tcp were measured in a 1952 L-head engine operating at 7.7/1 compression ratio. During deposit accumulation the commercial premium gasoline of Table 2 and an SAE 20W paraffinic-base oil were used.

Twelve measurements were used to establish autoignition tendency in each test. These tests were conducted in two cars operated with 0, 0.1, and 1.0 theory (Theoretical amount of phosphorus required to convert all of the lead to lead orthophosphate – $Pb_3(PO_4)_2$.) of tcp added to the tank fuel during the deposit buildup period. Results of these tests are given in Fig. 6. From this figure it is evident that 0.1 theory of tcp was ineffective in reducing autoignition tendency but that the addition of 1.0 theory, a quantity much larger than normally used, reduced the requirement for borderline autoignition by approximately three octane numbers. Autoignition failures were induced by moderate to heavy knocking in the cars using 1.0 theory tcp, but no knock was noticed prior to the occurrence of autoignition in the tests using untreated fuel. Thus, it would appear that retarding ignition timing when 1.0 theory tcp was used would increase the apparent benefit from this additive.

Representative photographs of piston tops and

Table 8 – Effects of Deposits from Leaded and Unleaded Fuels on Fuel Antiknock Quality for Acceptable Knock- and Autoignition-Free Operation

Properties of Base Fuel		
Specific Gravity at 60 F	0.738	
Reid Vapor Pressure, psi	9.3	
Gum, mg per 100 ml	0.4	
Sulfur, %	0.04	
ASTM Distillation, F		
Initial	102	
10%	124	
50%	204	
90%	316	
End Point	394	
Silica Gel Adsorption, volume %		
Paraffins	43	
Olefins	28	
Aromatics	29	
Car	A	B
Tel, ml per gal	0	2.0
Research Octane Number of Severity Reference Fuel for Acceptable Operation	92	95.5
Limiting Operating Condition	Knock	Autoignition

[8] U. S. Patent 2,427,173 (1947) to L. L. Withrow and U. S. Patent 2,405,560 (1946) to J. M. Campbell.

[9] "Service Experience with Tcp Fuel Additive," by C. R. Johnson and D. N. Harris. Presented at SAE Annual Meeting, Detroit, Jan. 16, 1953.

combustion chambers after operation on fuel containing 0 and 1.0 theory of tcp are shown in Fig. 7. Although not readily apparent in these pictures, it was found in the majority of instances that the deposits resulting from the use of 1.0 theory of the additive were a fluffy white and were less copious than those from operation with the untreated gasoline.

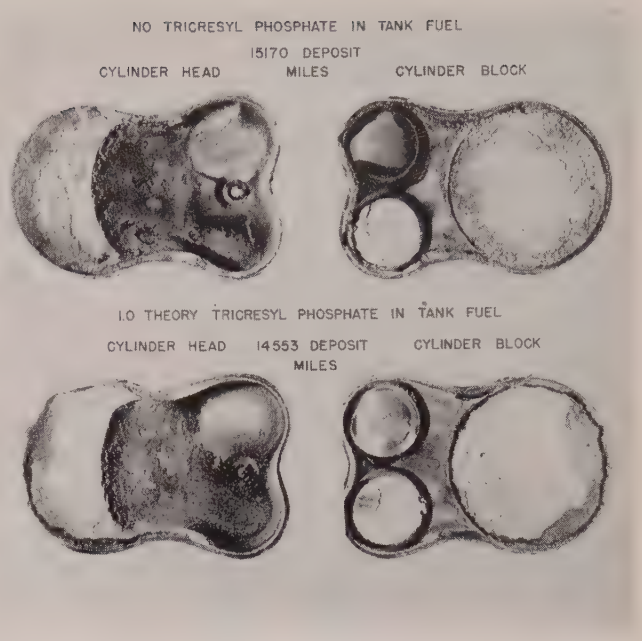

Fig. 7 – Combustion-chamber deposits with and without tricresyl phosphate. 1952 L-head engine; 7.7/1 compression ratio; leaded premium-grade tank fuel; paraffinic-base 20W oil

Summary

In this paper a number of factors related to the problem of autoignition as it occurs in automobiles on the road are discussed. This autoignition is characterized by erratic, sharp knocking or dull, thudding sounds, which persist after electrical ignition has been cut.

A test for measuring autoignition tendency is presented which uses the antiknock quality of members of a reference fuel series as the test parameter. It is shown that a number of observations, obtained during the process of deposit buildup, are required in order to eliminate the effects of random variation from the measurements.

Experiments are described which indicate that autoignition can be eliminated by increasing fuel antiknock quality, but that the Research octane number required may vary slightly depending on fuel composition. Limited evidence is presented indicating that aromatic constituents of gasoline are somewhat less effective in suppressing autoignition than are paraffin hydrocarbons.

It is shown that differences in autoignition tendency are observed when different fuels and lubricants are used during the deposit buildup period. In one series of tests a difference of seven octane numbers was observed between autoignition tendencies resulting from operation with different lubricants.

In another series of tests the Research octane number of the fuel required for satisfactory operation of a car run with gasoline containing 2.0 ml tel per gal was 3.5 octane numbers higher than that of a similar car operated on the same gasoline without tel. The limiting operating condition in the car using the leaded gasoline was autoignition; that of the car using unleaded fuel was knock. In appraising these results it should be recognized that the addition of tel to the base gasoline increased its Research octane rating by six octane numbers. Thus, there was a definite net gain despite the autoignition.

Experimental observations were used to show the effectiveness in reducing autoignition tendency of tricresyl phosphate additions to leaded gasoline during deposit accumulation. These tests indicated that addition of the quantity of this additive theoretically required to convert all of the lead in the fuel to lead phosphates lowered the Research octane number required to eliminate autoignition by approximately three octane numbers.

The authors wish to express appreciation to the management of the Research Laboratories Division of General Motors for permission to publish this paper, and to W. G. Agnew, F. W. Bowditch, J. D. Caplan and N. A. Hunstad for their help and criticism during its preparation.

Discussion of Preignition Symposium Papers

Out of the Frying Pan And Into the Fire

– J. B. Duckworth
Standard Oil Co. (Ind.)

AS Mr. Heron points out, preignition is affected by: engine characteristics, deposits, and fuel characteristics. The extent to which any one of these factors controls preignition is not fully understood. However, valve, spark-plug, and other combustion-chamber temperatures will doubtless continue to vary with engine design. As specific output is increased some part of the engine will become sufficiently hot to act as a source of ignition. Combustion-chamber deposits, including carbon, dirt, and ash resulting from the combustion of gasoline and lubricating oil, will reach high temperatures and may act to "preignite" the combustible mixture. In a very practical sense, two factors – engine de-

sign together with accumulated deposits – define the combustion problems imposed upon the fuel. For the past many years, anti-knock quality only has been the problem of major concern. However, as we make motor gasoline more resistant to knock or detonation it is reasonable to expect that some other form of distress will eventually show up. If complete elimination of the knock problem should result in a high incidence of preignition, we could well find ourselves out of the frying pan and in the fire.

The proof of the pudding – freedom from annoyance and possible engine failure accompanying preignition – is going to rest with both engine design and fuel quality. The very name "preignition" directly implies overheating of the environment in which the fuel is expected to work. Thus, we believe that responsibility for proper attention to this problem belongs to both the automotive and petroleum industries.

Heron Closure to Duckworth Discussion

MR. Duckworth does not mention his rather extensive personal experience with the destructive effects of preignition in aircraft engines during World War II. It is doubtless on the basis of this that he bases his view that eliminating knock and replacing it with preignition might well be jumping out of the frying pan into the fire. During the war he was particularly concerned with the relative resistance of very sensitive and relatively insensitive aviation fuels to preignition. As a result of listening to Mr. Duckworth's persuasive and dulcet tones I suddenly awakened to the fact that the Ethyl Corp. was committed to an extensive investigation of the preignition resistance of fuels. There was no question of the fact that the Air Force had an effective emissary.

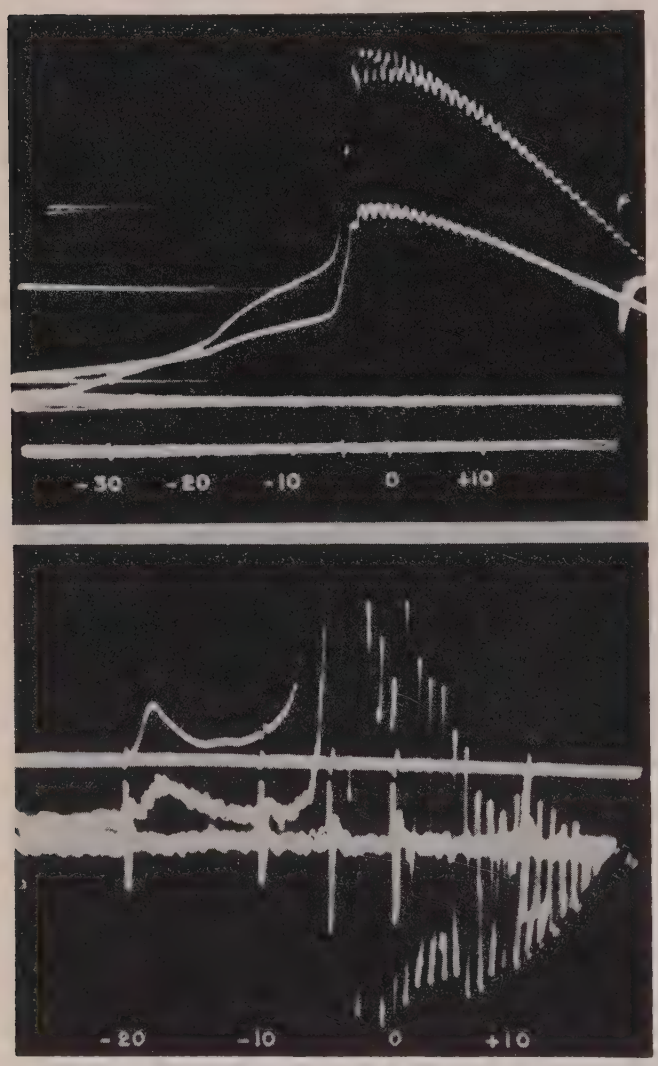

Fig. A – Pressure, radiation intensity, and rate of pressure rise versus crank angle during autoignition of lean n-heptane mixture

Autoignition Investigated In a Modified CFR Engine

– W. J. Levedahl
National Bureau of Standards

THE type of preignition which has been defined by Melby et al as "autoignition" has been the subject of an intensive investigation at the National Bureau of Standards. Much of this work has been carried out in a CFR engine which has been modified to permit the compression-ignition of a single, homogeneous, premixed fuel-air charge in the absence of burned residual gases, cylinder hot spots, and lubricating oil. Pressure, rate of change of pressure, and light emission are oscillographically recorded as functions of crank angle; a special sampling valve can be used to take samples from the reacting mixture during an interval of about 0.2 to 0.3 millisec.

Typical of the recordings made with the apparatus is Fig. A. The upper traces are pressure-time diagrams made on two different scales; they show the small pressure rise corresponding to the cool-flame reaction beginning at about 19 deg before top center and the hot-flame reaction becoming rapid at about 6 deg btc. The pressure vibrations responsible for audible knock are apparent in this diagram, although the "silent autoignition" mentioned by the authors has also been observed in this apparatus.

The bottom trace is the electronically differentiated pressure diagram, which provides a measure of the rate of heat release during the reaction. Just above this curve is a trace of similar shape showing light emission intensity, which in this case gives some evidence of a second cool flame at about 13 deg btc.

The lower line in Fig. B shows that fuel-air ratio has a very small effect on the temperature at which iodometrically detectable peroxides first appear in n-heptane mixtures. At a somewhat higher temperature the cool-flame reaction becomes exothermic, and at about 1090 F hot flame begins. Similar diagrams made with several pure hydrocarbons, isooctane-n-heptane blends, and oxygen-containing compounds show that, as octane number increases, the temperature at which peroxides and cool flames are detectable also increase, while the heat released during the cool flame decreases. Over a wide range of fuel air ratios, however, the temperatures at which hot flames first appear lie in the range 1100 F $\pm$ 50 for all aliphatic compounds investigated. Since the octane numbers and autoignition resistances of the fuels investigated vary widely, these properties seem to be functions of the extent of "self-heating" in the early stages of reaction. According to this view of the overall process, a very high-octane fuel must be heated by compression to nearly 1100 F, while a low-octane fuel may contribute several hundred degrees of the required temperature rise by its own early reactions. At this characteristic temperature there may be a sharp increase in the rate of formation of a common intermediate, or a reaction may occur which is common to all the fuels studied.

The variations in chemical composition and temperature of a lean n-hexane mixture during autoignition are shown in Fig. C. From the start of peroxidation until the beginning of intense cool flame no large chemical change can be detected.

During the cool-flame reaction, however, the hexane

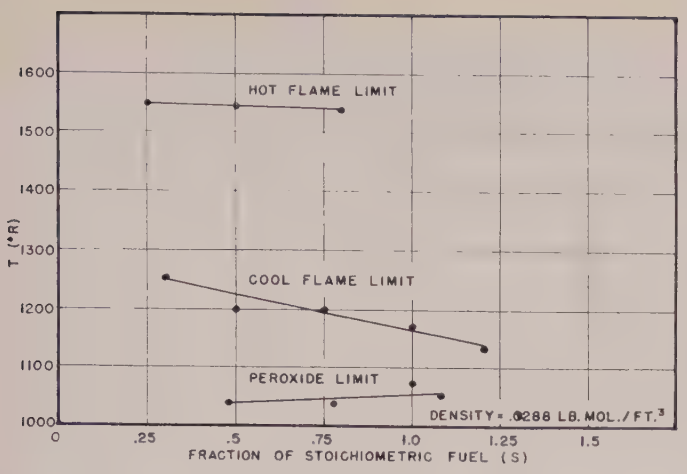

Fig. B – Reaction stage limits of n-heptane

begins to decompose into olefins, acetylene, formaldehyde, CO, CO_2, and water. By the time hot flames begin, only about one-third of the hexane has been decomposed, and this small amount of fuel has formed enough water and carbon oxides to cause a large rise in temperature. As hot flames begin, most hydrocarbons and formaldehyde are rapidly decomposed, while CO_2, CO, H_2O, hydrogen, and acetylene increase rapidly and small amounts of diolefins and benzene appear. The rapid rise of acetylene concentration is one of many pieces of information which tend to corroborate the results obtained in the past year by Dr. R. E. Ferguson of NBS, which suggest that acetylene is a very important intermediate in the combustion of all hydrocarbons.

The strong preignition tendency of benzene when exposed to hot, possibly catalytic, surfaces might be explained in terms of this theory by noting that benzene decomposed in the engine gives high yields of acetylene, diacetylene, and vinylacetylene. Aliphatics, on the other hand, are probably decomposed by hot surfaces in a stepwise process similar to that which occurs in pure compression ignition, so that their preignition tendencies might be expected to be proportional to their knocking tendencies.

Fuel Additives Help Control Deposit-Induced Ignition

– V. E. Yust
Shell Oil Co.

IT has been recognized for some time that combustion-chamber deposits and their effects on uncontrolled ignition have tended to limit the compression ratio of the L-head engine. The increased use of the overhead-valve engine has also focused attention on the problem, and it appears that in some cases compression ratio and maximum allowable power are being limited either by preignition or the more commonly observed wild ping.

Our laboratories have been studying combustion-chamber deposits for a number of years in aircraft and automotive equipment and have reported results on one phase of these studies in two papers.[a, b] The subject of deposit-induced ignition has likewise been under investigation, some of the highlights being discussed herewith.

In the course of our studies data were obtained in a bench-type apparatus wherein mixtures of lead salts and carbon were heated to a given temperature and the time required for ignition of the carbon was measured. Thus, it was possible to measure ignition delay for a number of lead salt-carbon mixtures over a wide temperature range. Some of the results showing the temperature at which there was a 10-sec ignition delay time are given in Fig. D. It is of interest that a composite sample of engine deposits exerted the greatest catalytic action on carbon, reducing the ignition temperature from 1200 F to about 750 F. Of the synthetic lead salt mixtures studied, the lead oxyhalide and oxide were the most active, being about equivalent to the composite engine deposits. The lead salt containing phosphorus, that is, the type obtained when using tricresyl phosphate in gasoline, exerted very little catalytic action. There is considerable difference in the composition of deposits in various locations in the combustion chamber, depending primarily on the temperature of the collecting surface. We have found that the activity of these deposits correlates very well with synthetic mixtures prepared in the laboratory.

Our study of spark-plug fouling and the role of combustion-chamber deposits led to the development and use of tricresyl phosphate as a supplementary lead scavenger for minimizing spark-plug fouling. Concurrently, the effects of tcp on preignition and wild ping were investigated. For the wild ping phase of the program we employed a naturally aspirated engine using the wild ping counter developed by Ethyl Corp. Some of the results are similar to those reported by Hirschler et al. Results of a typical test on leaded gasoline are shown in Fig. E. Both the octane requirement and wild ping level rose sharply in the early stages of the test and tended to stabilize thereafter.

We confirm the work done by Hirschler et al on the effect on wild ping exerted by operating conditions that are conducive to carbon formation. For example, mixture strength exerted a large influence on wild ping, as illus-

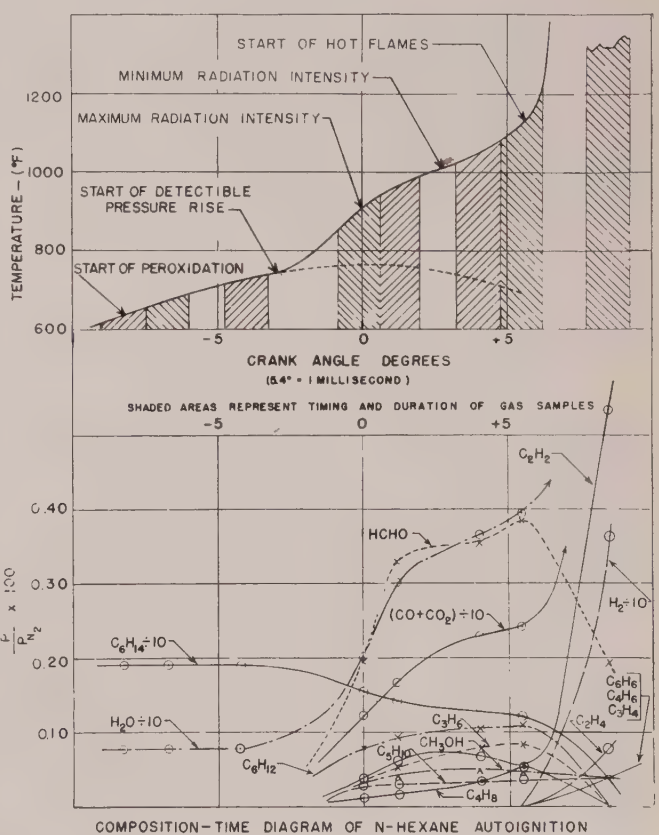

Fig. C – Temperature-time and composition-time diagrams of n-hexane autoignition

[a] "Aviation Spark-Plug Fouling – Its Cause and Control," by V. E. Yust and E. A. Droegemueller. Presented at SAE Annual Meeting, Detroit, Jan. 17, 1952

[b] See SAE Transactions, Vol. 61, 1953, pp. 3-13: "Spark-Plug Fouling Studies," by R. J. Greenshields.

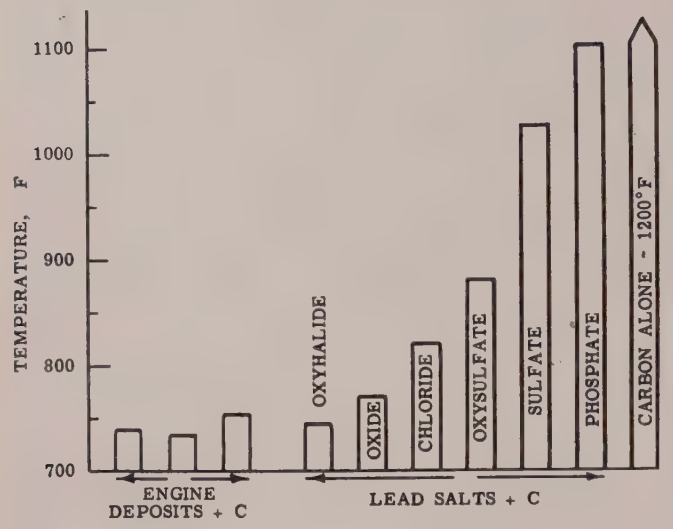

Fig. D – Temperatures for 10-sec ignition delay time

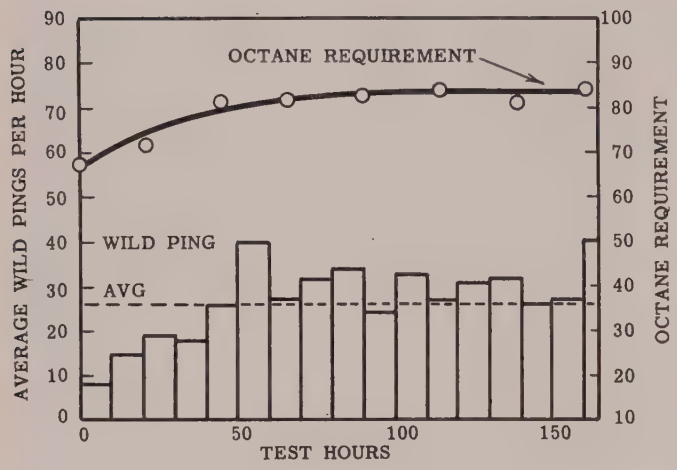

Fig E – Typical wild ping test results

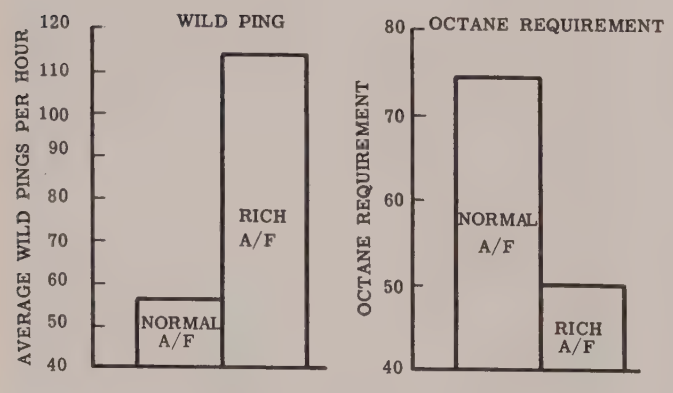

Fig. F – Effect of air/fuel ratio on wild ping

trated in Fig. F. Also, we confirm their data on the effect of oil consumption on wild ping.

The steady increase in compression ratio of automotive engines has compelled the fuel manufacturer to use more and more tetraethyl lead to satisfy the octane requirement of the newer model engines. This has greatly aggravated the wild ping problem, as shown in Fig. G. The relative number of wild pings shown is an average of the wild pings per hour over a 165-hr test period. It will be noted that an increase in lead level increased the relative number of wild pings from about 25 to 100. As a result of incorporating tcp into the highly leaded fuel, the wild ping was reduced to less than that obtained on the low lead fuel.

Fig. H shows the average results of tests on a 1951 Buick and a 1951 Cadillac over 300 hr of operation. With the conventional fuel, knock increased as deposits accumulated, resulting finally in preignition, which progressed to the severe state. At the 150-hr mark tcp fuel was introduced into the engine and, after a relatively few hours of further deposit accumulation, preignition ceased and the knock was thereby reduced. Thus, tcp also reduces preignition and wild ping in addition to alleviating the spark-plug fouling problem.

A rather large number of field tests have been run using tcp, and the results confirm laboratory experience except it would appear that the laboratory tests are probably on the severe side.

Again we concur with the authors that deposits from conventional leaded fuels greatly aggravate the wild ping problem. Furthermore, we believe that fuel additives can help to control deposit-induced ignition.

Presents Data on Effect Of Oils on Preignition

– F. I. L. Lawrence
Kendall Refining Co.

I SHOULD like to comment upon the section of the Williams-Landis paper concerning the autoignition tendencies of engines as affected by lubricant composition. We have been studying the effect of lubricants on preignition but most of our work has been done with multicylinder engines in the laboratory, where we have been somewhat successful. At this point, I would like to define preignition as ignition of the fuel-air mixture by the deposit at any point in the combustion chamber before the arrival of the normal flame front. This does not include detonation or knocking. Our preliminary efforts to study preignition in cars on the road have not been productive of much information to date. We believe that the techniques outlined by the authors will be very helpful to us in road test work.

I would like to present some data on the effect of oils on preignition. We have used in our work an experimental paraffinic oil similar to, but not identical with, oil A in Table 7, of the Williams-Landis paper. We will refer to this material as oil E in the data to be presented. We have also worked with oil C, the same oil as shown in Table 7 and which we will also call oil C.

The data shown in Fig. I were obtained in a 6-cyl, L-head engine with a compression ratio of 8.0/1.

In our procedure the distributor advance mechanism is locked and the spark set at top dead center. Oil and coolant temperatures are maintained at 160 F. The presence of preignition is determined through the use of ionization gaps installed in the end gas area of the combustion chamber in conjunction with an oscilloscope and allied electronic equipment. In this way, flame fronts originating from deposit-induced ignitions pass the ionization gap before the flame fronts associated with normal combustion and can readily be detected.

The engine is operated during test on a 4-min cycle as given in Table A.

The fuel used to operate the engine and build deposits

s a blend of platformate and thermally cracked gasoline with 3.0 cc of tel per gal to which 1% by weight of the oil under test is added.

Preignition requirements using reference fuels are taken at approximately 24-hr intervals until equilibrium of the combustion-chamber deposits has been reached. Reference fuels used in this determination are blends of commercial-type fuels (100 aviation, polymer, straight-run naphtha).

The broken lines in Fig. I indicate where there was an extended shutdown in that particular test. Good repeatabilty is indicated in the two tests on oil C. No preignition was observed with any of the oils until some deposit has been built up. The difference between oils E and C is about 10 Research octane numbers, whereas Messrs. Williams and Landis have reported 7 octane numbers between oil C and an oil similar to oil E in road test cars. This is relatively good agreement as to the magnitude of the effect.

It is interesting to observe that oil D has even less tendency to cause preignition than oil C. These two oils belong to the same family of experimental oils. It is, therefore, indicated that oils can be developed that contribute less to preignition than oil C.

We have also completed some laboratory and field detonation requirement increase tests on these same two oils, E and C. The laboratory data in Table B were obtained in a 6-cyl, L-head engine with a compression ratio of 7.0/1. In this procedure, automatic spark advance is used. The fuel employed is a blend of platformate and thermally cracked gasoline with 3.0 cc of tel per gal added. Oil and coolant temperatures are maintained at 160 F.

Upon completion of a breakin run, the combustion chambers are cleaned and a borderline detonation requirement taken under each of the conditions given in Table C.

Reference fuels used are blends of commercial-type gasolines (premium gasoline and straight-run gasoline).

Under test, the engine is operated on a light-duty 4-min cycle as given in Table D.

The part-throttle requirement is determined at intervals of approximately 24 hr until the combustion-chamber deposits reach equilibrium or about 110 hr. At that time, both part- and full-throttle determinations are made. The engine is cycled for an additional 10 hr to replace any deposits disturbed during the final determination, after which the combustion-chamber deposits are removed and weighed. A final clean determination is then made to ascertain any changes in engine condition.

The results in Table B show that the Research octane-number increases in engines lubricated with oil C were less than half the increases for oil E for both full- and part-throttle conditions. Likewise, the weight of deposit removed from combustion chambers where oil C was used was less than half that for oil E.

The results of laboratory tests just described have been confirmed in cars on the road. However, the differences tend to become smaller in these uncontrolled field test cars. Others working with combustion-chamber problems have had this same experience. These field test data are shown in Table E. Cars A and B are 1952 models with a 7.5/1 compression ratio and have overhead valves. Car C is a 1951 model of the same make. Car D is a 1953 model powered by an L-head engine at 7.7/1 compression ratio. Deposits in all cars were accumulated to equilibrium, which was reached at about 6000 miles. A clean requirement was first obtained. Detonation requirements were then run every 2000 miles and a final clean requirement was also obtained.

Detonation requirements are determined over a level section of highway. Full-throttle accelerations in both third and fourth gear are made using primary reference fuels. The requirement of the car is given as that octane fuel which allows no more than borderline knock during any part of the acceleration.

The increase in Research octane number of the reference fuels required for borderline detonation varies between cars lubricated with the same test oils. This might be

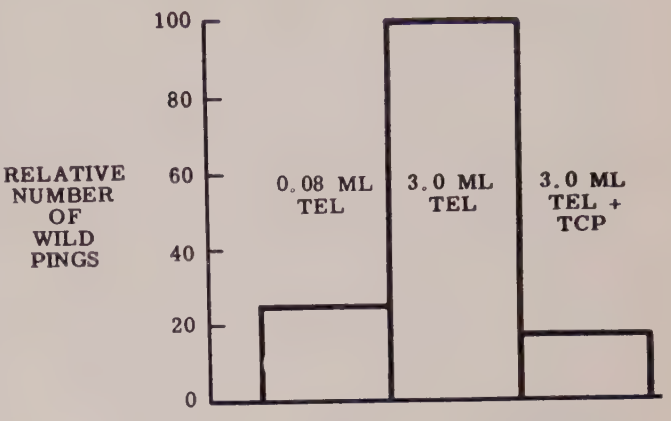

Fig. G – Effects of tel and tcp on wild ping – single-cylinder cyclic test

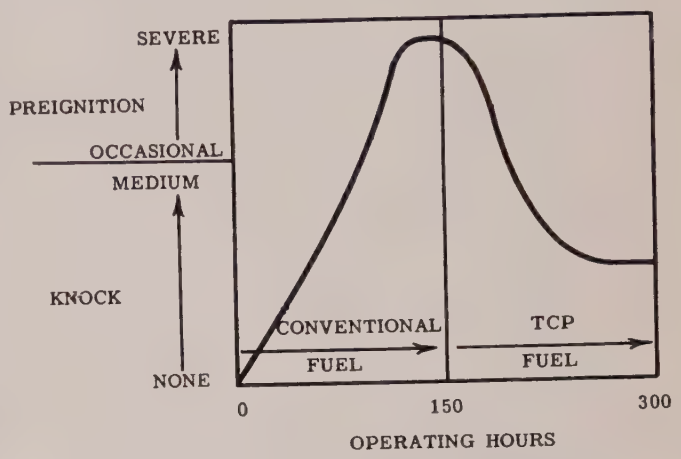

Fig. H – Effect of deposits on knock and preignition in full-scale automotive engines

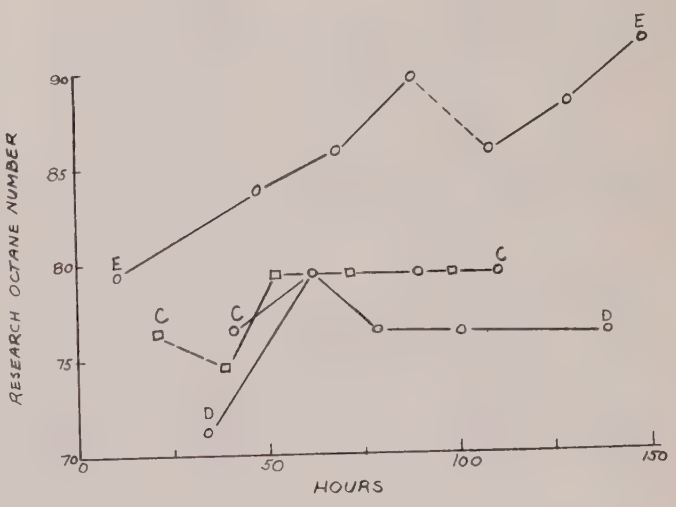

Fig. I – Laboratory preignition tests – commercial-type blend

Table A – Preignition Requirement Test Cycle

Time, min	1	3
Speed, rpm	700	1500
Load, bhp	0	30

Table B – Detonation Requirement Increase – Laboratory Tests

Oil	E	C
No. of Tests	3	7
Full Throttle	13.5	6.0
Part Throttle	24.1	10.9
Deposits, g	85	34

Table C – Detonation Requirement Conditions

	Part Throttle	Full Throttle
Speed, rpm	1500	1500
Load, bhp	37.5	Maximum
Spark Setting, deg btc	7	7

Table D – Detonation Requirement Increase Cycle Conditions

Time, min	1	3
Speed, rpm	600	2000
Load, bhp	0	10
Spark Setting, deg	2 atc	28 btc

Table E – Detonation Requirement Increase – Field Tests

Oil	E	C
Car A	13.5	8.5
	10.0	8.0
Car B	15.0	6.0
	16.0	
Car C	18.0	14.0
Car D	16.0	12.0

Table F – Predicted and Actual Magnitude of Peak Pressures, Time of Spark Discharge, and Time of Occurrence of First Ionization Indication

Time of Spark Discharge, deg btc	Predicted[a] Magnitude of Peak Pressure Based on Time of Spark Discharge, psi	Time of First Ionization, deg btc	Predicted[a] Magnitude of Peak Pressure Based on Time of First Ionization, psi	Actual Magnitude of Peak Pressure, psi
0	300	17	580	298
8	455	36	710	389
15	555	23.5	640	530
23	630	40	725	620

[a] Predicted from the known variation of magnitude of peak pressure with time of ignition from a normal-burning cylinder of the same engine under the same conditions.

Table G – Predicted and Actual Time of Peak Pressures, Time of Spark Discharge, and Time of Occurrence of First Ionization Indication

Time of Spark Discharge, deg btc	Predicted[a] Time of Peak Pressure Based on Time of Spark Discharge, deg atc	Time of First Ionization, deg btc	Predicted[a] Time of Peak Pressure Based on Time of First Ionization, deg atc	Actual Time of Peak Pressure, deg atc
23	12	40	1	15
15	19	23.5	11	15.5
8	28	36	3	24.7
0	42	17	17	42

[a] Predicted from the known variation of time of peak pressure with time of ignition from a normal-burning cylinder of the same engine under the same conditions.

expected as the cars were in different types of service. However, oil C shows up better in each of the test cars.

In conclusion, it has been shown that lubricating oil can have a marked effect, not only on the autoignition or preignition characteristics of combustion-chamber deposits but also on the detonation or knocking characteristics. The results obtained with a new family of experimental oils has been briefly touched upon. It should be mentioned at this point, that patent rights in the development of lubricants of this type have been reserved.

Peculiar Results Reported With Ionization Gap Circuit

– F. W. Bowditch

GM Research Laboratories Division

IN connection with the use of the spark-plug ionization gap circuit originated by Messrs. Winch and Mayes we have found some rather peculiar results. In our application of this circuit to cars on the road we have found advanced preigniting indications from the ionization traces which were different in that they were: (1) very consistent, (2) fuel insensitive, and (3) not accompanied by any abnormal engine noises. In some cars this early ionization would appear in some cylinders at or above a certain engine speed regardless of how the car was driven. Once the early ionization appeared, the time of occurrence in crank angle degrees of the first sign of ionization was very consistent. Changes in fuel antiknock quality and composition and changes in spark plugs altered neither the time of occurrence of the first ionization nor the engine speed at which it first appeared.

In order to verify this indication of silent preignition, pressure indicators were mounted in two cylinders of an eight-cyl L-head engine; one indicator was mounted in the cylinder in which the ionization trace indicated the most advanced preignition, and the other pressure indicator was placed in a cylinder which showed no indications of preignition. Fig. J shows a representative record obtained from a dual-beam oscilloscope in which the upper trace is a pressure-time record and the bottom trace is the ionization records for five of the eight cylinders. The time scale for both records is the same.

As can be seen, spark discharge was occurring at 8 deg btc. In cylinder 7, the cylinder for which both the ionization trace and the pressure record appear, very definite signs of early ionization appear, the first sign of this early ionization appearing at 36 deg btc or 28 deg before the spark fired. Peak pressure for this explosion occurred at 25 deg atc and was 390 psi.

If advanced preignition was occurring, the magnitude and time of occurrence of peak pressure should certainly be altered from a normal burning explosion operating under the same engine conditions. The variation of magnitude of peak pressure with time of ignition shown in Fig. K was obtained from records in the cylinder which showed no signs of preignition for an engine speed of 1600 rpm and an intake manifold vacuum of 0.6 in. Hg.

From records similar to that shown in Fig. J taken under the same conditions of engine speed and vacuum, magnitudes of peak pressure were predicted from this curve based on the time of spark discharge and the time at which the first ionization indication appeared. These predicted values were then compared with the actual values of peak pressure obtained for these early ionization pressure records.

Table F presents the tabulated values of predicted and actual peak pressures, time of spark discharge, and time of occurrence of the first ionization indication of over 25 records for each condition. The predicted values of peak pressure based on the time of first ionization are the smallest values possible. If preignition was occurring, the probability is that one or both of the following would

occur: (1) There would be more than one center of ignition, and (2) the center of ignition of the flame front that caused the early ionization would not be at the spark plug and, therefore, would occur earlier than the indication on the ionization trace. Both of these effects would tend to make peak pressure larger. As can be seen from Table F, the predicted values of peak pressure based on the time of spark discharge agree fairly well with the actual values, whereas the other predicted values based on the early ionization are definitely too high. From this analysis it appears that, for the cases shown, the phenomenon which caused the occurrence of the early ionization did not affect the magnitude of peak pressure.

A similar analysis of these same data may be made based on the known fact that the time of occurrence of peak pressure is influenced by the occurrence of preignition. Fig. L shows the variation of the time of peak pressure with time of ignition in the same cylinder which showed no signs of preignition. Predicted values of time of peak pressure were obtained based again on both the time of spark discharge and the time the first indication of ionization appeared. A comparison of the actual time of peak pressure and the predicted values based on the time of spark discharge and early ionization of the same records will indicate whether or not preignition occurred in these cases.

In Table G, again each value of spark discharge and time of first ionization are average values of at least 25 records. The predicted time of peak pressure based on the time of first ionization is the latest possible time in crank angle degrees since, again, if autoignition were occurring, more than one source of ignition would probably be present, and the early ionization combustion source would probably not be at the spark plug, both effects tending to make the predicted time of peak pressure based on the time of first ionization earlier than the time shown in Table G. Again, it appears that for these cases, the phenomenon which caused the early ionization had little, if any, effect on the time of occurrence of peak pressure.

Apparently, in the cases cited, preignition was **not** occurring. A full explanation of this early ionization in some cylinders has not been established.

Winch Closure to Bowditch Discussion

DR. Bowditch is to be congratulated on his very thorough and interesting proof of the absence of preignition in the particular cylinder he evaluated. Through examination of the ionization trace he has shown as Fig. J, I would agree with his conclusion that cylinder No. 7 was not preigniting.

The procedure used in accumulating the data presented in the paper does require considerable experience and interpretation on the part of the operator. This is a disadvantage that is common to almost all test techniques and procedures used in testing automotive engines. The phenomena Dr. Bowditch has shown in Fig. J and described in his discussion were also observed in some of our work. However, the consistency of occurrence, the lack of sensitivity to fuel changes, and the severe proportions it would assume without causing any abnormal engine noise, roughness, or run-on without electrical ignition made us conclude that this must represent an ignition system malfunction. Another laboratory reported the same difficulty. It found that this signal could be eliminated by inserting a series gap in the high tension wire from the coil to the distributor. We have since had an opportunity to confirm this. The theory is that a small amount of charge is stored in the secondary circuit of the coil from the preceding spark discharge. If this residual charge is strong enough, or if the distributor rotor is so timed that it reaches the next distributor segment too soon, it will jump from the rotor to the cap thereby causing the pattern reported by Dr. Bowditch.

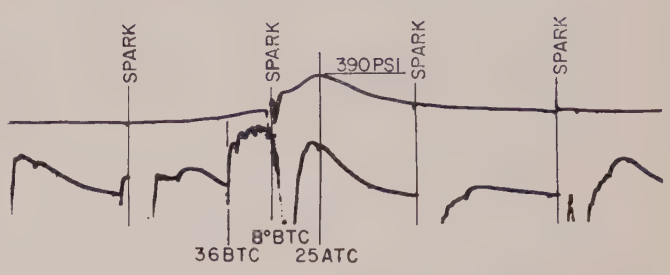

Fig. J – Upper trace is pressure-time record, lower trace is ionization record for five of eight cylinders

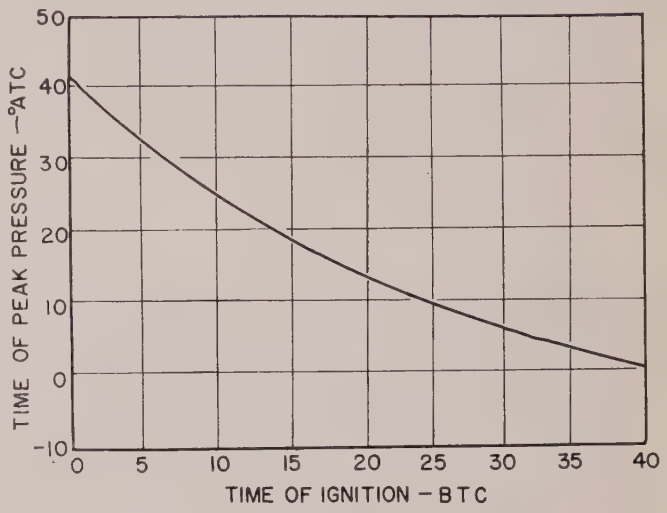

Fig. K – Effect of time of ignition on time of occurrence of peak pressure

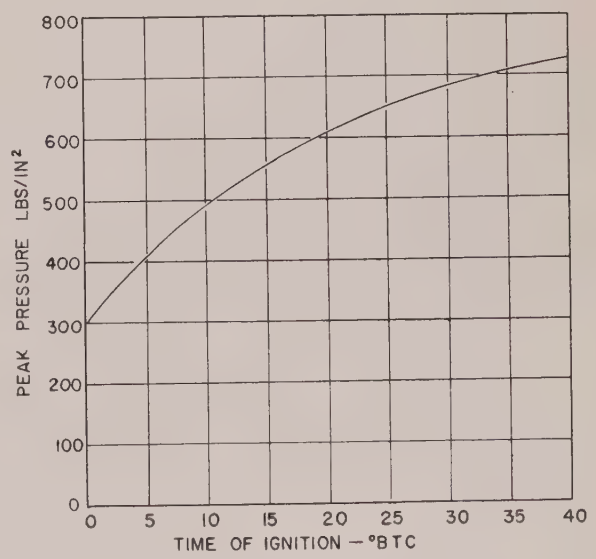

Fig. L – Effect of time of ignition on magnitude of peak pressure

The Luster

ACTUALLY whether we realize it or not, so far as the interior decoration of automobiles is concerned, we are in an era of a luster revolution, and this luster revolution is made possible by the availability of new manmade fibers which are essentially ideal for use in automobile upholstery fabrics. These new manmade fibers, because of their inherent worth, have so increased the prestige of synthetic fibers that they have served to emphasize the basic values of some of the older manmade fibers which had not been heretofore fully appreciated, and to that end served to popularize manmade fibers in general.

Some of the manmade fibers have been in rather common use in many textile fields for quite a few years, but their extensive use by automobile trim stylists and automobile engineers has been of comparatively recent origin.

I can recall very vividly some years ago that I went to Detroit to visit one of my friends with the express purpose of trying to interest him in a new yarn and in a new fabric which we thought

PROGRESS made in the development of manmade fibers is precipitating a revolution in the design of automobile upholstery fabrics, according to the author. In various combinations with each other, with the older manmade fibers, and with natural fibers, they are capable of producing fabrics of increased decorative value together with a serviceability heretofore unthought of.

The author discusses the properties of some of these synthetics. He points out that none of them can be called the universal fiber. Each has individual characteristics of outstanding merit, but not one of them alone is suitable, for one reason or another, to all textile applications. On the other hand, combinations of these fibers yield properties not possessed by the individual fibers themselves.

The Author

W. F. BIRD is executive vice-president of Collins and Aikman Corp. He has been with the firm since 1927. After graduating from Northeastern University as a chemical engineer, Mr. Bird became a field engineer for a firm of Boston consulting engineers. In this work he was concerned with the problems of waste disposal and recovery from wool scouring. Mr. Bird is well known for his many patents for spinning, weaving, dyeing, and finishing

Revolution

W. F. Bird, Collins and Aikman Corp.

Paper presented at SAE National Passenger-Car, Body, and Materials Mtg., Detroit, March 4, 1953.

had great possibilities. This new fabric contained a fairly small percentage of one of the accepted manmade fibers of that day. My friend's rejection of the fabric was immediate and was based on the fact that the introduction of the manmade fiber gave the material a slight luster which was extremely objectionable to him and his associates because this luster detracted from the soft, dull, rich appearance of the automobile interior, but which I think could be more aptly portrayed as taking away from the quiet funereal-like dignity which was the motif of that day.

Consider the change in thinking of automobile interior stylists from that period to the present day, where luster, life, and color obtained by proper selection in combinations of manmade fibers in bolster, seat covering, and sidewall are the objectives. There never has been a period in the building of automobiles when the ultimate user has received as much serviceability and style appeal from the fabrics used in upholstering and decorating the interior of these automobiles as he does at the present time.

Through the years the three principal fibers that have been used in the manufacture of cloth for automotive upholstery have been cotton, wool, and mohair.

Fairly recently, and to a somewhat limited extent, acetate and viscose were added to this list, and more recently have been added Nylon and Orlon. Additional new manmade fibers are appearing experimentally in increasing numbers. More than 70 entirely new fibers have been produced in some small laboratory or pilot plant quantities. The eight synthetic fibers which are now offered, or soon will be offered in commercial quantities include Dacron, Dynel, Acrilan, X-51, Perlon, Kuralon, Saran, and Vicara.

Other fibers now in the laboratory development stage are due to make their appearance in the next few years. Before these fibers will be accepted, they will have to prove their worth. None will be adopted merely because they are new or have been described by publicity men as "wonder fibers" or "miracle fibers." Some will not be successful, but you can be sure that each one will stand or fall on its own merits.

There is no universal fiber which answers the purpose of the automotive stylist and the textile engineer in all end purposes. Each of the fibers, including the natural fibers, has individual characteristics of outstanding merit, but not one of them alone is suitable, for one reason or another, to all textile applications. In addition to the individual merit of any particular fiber, textile engineers have found that combinations of these fibers yield properties not possessed by the individual fibers themselves.

The number of possible mathematical combinations of manmade fibers with manmade fibers, or manmade fibers with natural fibers, becomes very large indeed. Thus if the five older fibers, namely cotton, wool, mohair, acetate, and viscose, plus the two newer ones in general use, plus the eight new synthetics shortly to come into quantity production are combined as 2-fiber mixtures, these

15 fibers would yield 105 different possible combinations. If the enthusiastic textile engineer or automobile body stylist is not satisfied with 2-way combinations and wants to try for further innovations, he can obtain 455 3-way combinations; 1365 4-way combinations; or over 3000 5-way combinations.

When it is remembered that, in addition to this, there must be considered the various forms in which these synthetics are available, the several methods of preparing yarns, the possibility of varying the percentages of fibers within the yarns, the many fabric constructions and methods of dyeing and finishing, the total possible combinations and variations with which a stylist can work become practically infinite.

This complex problem obviously requires that the stylist and fabric manufacturer be intimately acquainted with the merits and faults of the individual fibers if they are to work with them intelligently to build into a fabric the particular properties which are required. Therefore, let us look briefly at the outstanding characteristics of some of these individual fibers.

Nylon and Orlon

Nylon is available both as staple and filament. This polyamid fiber is notable for its high tensile strength and its outstanding resistance to abrasion. It is highly elastic and recovers well from stretching. It does not shrink. It washes easily and dries quickly. It has excellent resistance to mildew and insects. It shows good resistance to burning. Because Nylon has low water absorption, good fastness to light fading has been a problem, but the development of new special dyestuffs is markedly improving this fault. The price is high but stable. The supply of fiber is fairly good and is rising at a moderate rate. It is now being widely used in automotive fabrics.

Orlon is the one acrylic fiber derived wholly from acrylonitrile. It has superior resistance to sunlight and excellent bulking power. It exhibits good wrinkle resistance and crease recovery. It has satisfactory wear resistance. Orlon is easy to wash and dries quickly for the reason that it is even more hydrophobic than Nylon. It does not shrink, has good draping quality, and is warm to the touch. It is difficult to dye, and this property has been used to achieve pattern effects. The price is high but stable. The supply is moderate and rising.

Dacron

Dacron is a polyester fiber which is characterized by superior wrinkle resistance, crease recovery, and unusual resilience. It has very little stretch and rates next to Nylon in tensile strength. It possesses excellent resistance to abrasion and to other wear factors. It is degraded only very slightly by exposure to sunlight. It has good resistance to mildew and is not attacked by moths. It has good draping quality and is warm to touch. It washes and dries easily for the reason that its water absorption is less than that of all other fibers. Because of this it is difficult to dye. However, a new class of dyestuffs has been found which will produce shades comparable with the best vats on cotton. The price is high but stable. The supply is small and it will not be available in even moderate quantities before early 1954.

Dynel

Dynel is classed as an acrylic fiber. Because of its 60% vinyl chloride content it has superior resistance to burning and is little affected by acids and other chemicals. It is characterized by good bulking power, good wrinkle resistance and crease recovery, good drapability, and moderate resistance to wear. It is warm to touch, washes easily, and does not shrink on contact with water. It is, however, susceptible to shrinkage with heat and must be handled through textile processing at relatively low temperatures. Shades of modern depth and fair light fastness may be achieved by conventional means, but heavy shades require special dyeing techniques. The price is moderate and stable. The supply is only fair, but is increasing. A new modification of Dynel with properties more like Orlon will soon be available. This new fiber will be much more heat stable.

Acrilan

Acrilan is another acrylic fiber. Like Orlon and Dynel it has superior bulking power, good wrinkle resistance and crease recovery. It has fair wearing quality and good resistance to sunlight, acids, insects, and mildew. It is easy to wash, dries quickly, and does not shrink. It has good draping quality, is warm to touch, and has fair resilience. Although it dyes more readily than Orlon, there is still considerable to be desired. Acrilan is constantly being improved both in its physical characteristics and in its dyeability. The price is high but stable. The supply is small, but if and when the properties become stabilized, the supply will increase rapidly. It is available only as staple.

X-51

X-51 is the acrylic with the best dyeing characteristics – conventional methods can be used. This fiber has good bulking characteristics, and remarkable wrinkle resistance and crease retention even when wet. Its wear resistance is good. It has good drapability and excellent resistance to attack by

insects, mildew, and chemicals. It has superior soil resistance and washes and dry cleans readily. Because of its low water absorption, it dries quickly. It is about like Orlon and Acrilan in resilience, but is a somewhat softer fiber. It is warm to touch. It is still in the pilot plant stage and hence the supply is very small. It is expected to be in production in about two years. The price is high but stable.

Perlon

Perlon is a polyamide fiber similar in many respects to Nylon. It is at present imported but at least two domestic manufacturers will produce it within a year and a half. It has a high tensile strength, resilience, and melting point which very nearly parallel those of American Nylon. It is competitive in price with Nylon.

Kuralon

Kuralon is a Japanese fiber derived from polyvinyl alcohol and has no available domestically produced counterpart. Although it is more like cotton than wool, it is remarkable for its resistance to abrasion. It is available only as semidull staple and fabrics made from 100% Kuralon lack character. It dyes readily with most classes of dyestuffs, but light fastness is, in general, not good. The present price is moderate, but the supply is small.

Saran

Saran is a polyvinylidene chloride fiber and is mostly sold as monofilaments. It is also marketed under the name of Velon. It is made in small quantities as 10d and 22d/f staple and tow. It has been used alone and in conjunction with wool in the manufacture of carpets. The monofilaments are used for seat covers. The high chlorine content makes this fiber practically flameproof. Because of the extreme difficulty in dyeing, both the monofilaments and multifilaments are usually "dope dyed" – that is, color bodies are added to the hot melt before the fiber is extruded. The cost per pound is moderate, but the density is high so that the coverage per pound is less than that of other fibers.

Vicara

Vicara is derived from Zein, the protein of corn. It is superior in softness, hand and draping quality, and dyes well in a wide color range. The newer forms have good dimensional stability, even though the moisture absorption is high. It blends well with other fibers and usually improves the spinnability, adds softness, and improves the hand. Unfortunately the wet strength is low, but this can be offset by blending with the stronger fibers such as Nylon, Perlon, or Dacron. The price is moderate and stable. The supply is moderate and rising rapidly.

Five Common Fibers

The properties of the five more common fibers are much more familiar, so that only a very brief summary will be given.

Cotton is the principal textile fiber in use, and its importance cannot be overstressed. Its good strength both wet and dry, excellent abrasion resistance, and outstanding flexibility are well known. It dyes readily and can be spun to fine counts. It is widely used for backing and filling yarns in automotive upholstery fabrics. Because of its easy soilability and its lack of resilience, it is not much used in face or decoration yarns in such fabrics.

Viscose is practically pure cellulose regenerated from low-grade cotton or wood pulp. It is available both as staple and filament and is marketed as dull, semidull, and bright forms. It is probably the most important synthetic fiber because of its excellent physical properties, large volume, and low price. It is like cotton in many of its properties. However, its wet strength is less than that of cotton. Like cotton, it soils readily and has poor resilience.

Acetate is also derived from cellulose. Unlike cotton and viscose, it is not readily dyed and requires special acetate dyestuffs to color it. This property is sometimes used to produce special color effects when the fiber is left undyed or only slightly stained. Acetate has somewhat better resilience than cotton or viscose, but its tensile strength is somewhat less.

Wool is still regarded as one of the most versatile fibers and because we probably know more about its basic characteristics and processing idiosyncrasies, wool is generally used as a standard fiber against which synthetic fibers are compared. Its excellent wearability, unusual dyeability, high resilience, recoverability, and good moisture absorption are a few of the properties which have caused it to be so highly regarded and generally used.

Mohair is a type of super wool known for its excellent resilience and high luster. Its other properties are much like those of wool.

As I have mentioned previously, there are hundreds of thousands of possible combinations of the available textile natural and manmade fibers. The textile engineer, working in close cooperation with automotive body stylists and engineers, has explored many of these new combinations in search of improved results, exciting new effects, and increased value for the ultimate automobile purchaser. The results have been startling to the point of precipitating a revolution in the design of fabrics, their decorative value, and their intrinsic worth. The future is going to be no less exciting because the possibilities of increasing the decora-

tive value together with serviceability heretofore unthought of are endless.

The most popular and magical of the manmade fibers for automobile interior decoration today is Nylon and there is very good reason for this because Nylon has most of the characteristics which, either used alone or in blends with other fibers, increase the wear, beauty, life, and decorative value of the upholstery fabric. As a matter of fact, the intrinsic value of Nylon is probably more responsible for the luster revolution than any other single item.

I would like to discuss briefly the form in which these fibers are produced and available to the textile industry and exactly how textile engineers make use of them for various effects.

Most manmade fibers are produced in two forms, one is staple and the other is continuous filament. Both of these forms are produced in varying deniers or sizes of individual filaments, together with varying numbers of these individual filaments grouped together in the case of the so-called filament form.

Filament yarns, as the name designates, are long lengths of continuous strands of the particular manmade fiber and are used by textile engineers generally for the purpose of imparting luster, brilliance, smoothness, and sleekness to a fabric. The textile engineer does not have to make yarn out of filament because it is already in a continuous form, as I have indicated. However, there is considerable work that has to be done on this fiber as it comes from the manufacturer before it is suitable for textile purposes.

In the first place, the yarn is manufactured originally by extrusion through a nozzle somewhat like the top of a salt-cellar, this nozzle having had made in it microscopic holes the size of the desired individual filaments and of the desired number to give the denier thread wanted. This extruded group of individual filaments has little or no twist and the individual filaments, as can be appreciated, are extremely delicate. Before they are used by the textile engineer they must be put in additional machinery, which imparts the desired turns of twist either to the original group of fibers as described above or to any number of these grouped together to give the desired result. Varying degrees of twist in a filament yarn are used by textile engineers to give varying decorative effects in the resulting fabric.

In general, increasing the twist decreases the original luster of the fiber. This process of putting twist into filament yarn is called "throwing," just why I could never tell you, and from some of the troubles that the textile industry experiences with so-called throwsters I wish sometimes they would throw it a long, long way away!

After the filament is thrown, plied, and twisted to the desired size, and so on, many of the fibers, such as Nylon and Orlon, have to be thermal set to prevent residual shrinkage later on in the processing. To accomplish this it has to be wound on specially constructed packages and put into ovens where the temperature and humidity are carefully controlled. The material is literally "cooked" for a very definite period of time. The purpose of this is to quiet their internal frustrations and to line up the molecules of each individual fiber in such a manner that they will not give difficulty later in manufacturing by having too much or too little stretch or shrinkage.

Actually, this is a very delicate operation, and sometimes, for particular purposes, the outside of the package may not get exactly the same amount of "cooking" as the inside. When this results, the package has to be removed from the oven and rewound with the outside in and the inside out and cooked again. No new bride ever watched her first cake any more carefully than the operator watches the conditions which control the setting of this material.

When you hear of streakiness in fabrics made from manmade fibers, many times the reason for this streakiness can be found in the improper thermal setting of the original fibers. The variations in the shrinkage characteristic of adjacent threads cause differences in light reflection, which in turn give the appearance of streaks.

If I may go back to the twisting and plying operation for a moment, I would like to emphasize the importance of very careful control in this operation. Variations in the twist of a fraction of a turn plus or minus the standard will also give a most horrible streaky effect in the fabric. The fact that some of the throwsters have not been conscious of the very careful control that is necessary when these fibers are used in the newer constructions for upholstery fabrics leads me to the remark which I previously made about throwsters throwing themselves and their product a long way away.

Another very sad aspect is that the textile manufacturer cannot detect these defects in the yarn or even in the greige fabric. It is not until the fabric is finally dyed and finished that they are evident and of course then it is too late to do much about it.

Another problem which faces the textile manufacturer when using manmade fibers is what we would ordinarily call a "change of batch," but which the manufacturers of these manmade fibers for some reason or other call a "new merge."

In earlier discussions on this subject I emphasized the differences caused by having to use the natural fibers such as cotton or wool, grown under various conditions of climate and soil, and in the case of sheep and goats (from which come wool and mohair), even a change in the materials they feed on. At that time, being less experienced in the wonders of chemical wizardry, I hoped for the day when materials could be produced synthetically, and their every stage carefully controlled by learned scientists, to the end that the finished

product would never vary in any of its basic characteristics. This would eliminate some of the most exasperating problems the textile engineer has to cope with, but with all due respect to the vast ability of the fine men and women who have developed these manmade fibers, no sheep born in the depths of darkest Africa compared with his brother from Iceland could ever be as different in their processing characteristics and their dye absorption index as varying merges of manmade fibers! We hope this is one thing that our learned friends who make these manmade fibers will work on diligently to overcome.

Staple Fiber

Staple fiber is the other form in which manmade fibers are produced. Staple is essentially filament which has been cut to any desired length. Like filament it is produced in varying degrees of luster such as bright, semidull, and dull, and in addition is usually crimped to give the individual fibers certain clinging characteristics which assist in the

Although staple and spun yarns made from this staple are made out of material which in many cases is exactly the same as the original filament, the characteristics of the yarns made from these cut fibers and the characteristics of the fabrics in which they are ultimately used are quite different. These characteristics differ mainly in that yarns spun from staple are less brilliant, more lofty or less sleek, and impart to a fabric a much fuller, softer, kindlier handle.

Oftentimes the textile engineer will use combinations of filament and spun either plied together, laid adjacently in fabrics, or in other evident combinations, because the differences in appearance make very pleasing design and decorative effects in the finished fabrics.

The processing of staple into yarns is a most interesting process.

To prepare staple fiber for further processing, the textile engineer takes the compressed bales of cut fiber and deliberately puts them through machinery designed to separate each individual fiber from the other. To all intents and purposes, we literally scatter the fibers to the four winds and then gather them together again and attempt to take fiber by fiber and put them together in a continuous, uniform strand. To the uninitiated this would seem to be a senseless way of doing things, and perhaps it is. Perhaps the textile engineer can't see the forest for the trees.

I have spent years in inventing and perfecting a system which throws these fibers further and gathers them together again in what I believe is a superior manner to make an ultimate yarn at less cost and with superior characteristics. The Government patent office must be inured to crazy textile engineers because they have granted me a patent on this process.

After the staple is cut up and put through beaters to separate each fiber from the other, it is put through a card, which is simply a system of revolving cylinders covered with wire bent at definite angles, the various cylinders going at carefully predetermined speeds so as to pick up the fibers one from the other and join them together at the end in a continuous band or sliver, as we call it. The strand has no strength; the fibers are positioned in every conceivable way, but at least it is a continuous strand of material from this machine, and subsequent processes are designed solely to draw this strand gently down and down in size until it is small enough to meet the particular requirements for which it is to be used, and in this drawing or "drafting" as it is called in the textile industry, the individual fibers assume more and more of an end to end or parallelization form.

Various ingenious systems of drafting have been devised to hold one end of the fiber while the other fibers are drawn past it, which all has a tendency to parallelize one fiber to another. In some instances where the strand and the fibers are particularly tangled or particularly difficult to manipulate into a paralleled position, the whole strand is put through another ingenious machine which is actually a comb, and has the same effect on the fibers being processed as a comb does when you draw it through your hair.

I won't go into any further detail as to just how these things are accomplished, because such a discussion is a long subject in itself. I think it is sufficient to say that, depending on the degree of efficiency of the equipment and of the people tending the equipment, the final yarn made after scattering the individual fibers to the four winds, gathering them together, and processing them as I have described, comes out in an amazingly uniform, continuous strand which can be closely regulated as to weight and size.

The progress made in the development of superior manmade fibers has been most impressive. So impressive, indeed, that it is precipitating the revolution in the design of the automobile upholstery fabrics that I have spoken of.

Generally speaking, the know-how of the manufacturer and the textile engineer has kept pace with the enormous demand from the ultimate user for fabrics incorporating the superior serviceability and decorative effects to be obtained when these manmade fibers are used.

In certain instances, however, emergencies have arisen where unforeseen difficulties are encountered. In such instances, of course, great concern exists until these emergencies are overcome. The old ones have been overcome and new ones will be overcome. I know of no way to learn to do a job except to do it! If we make progress cautiously and intelligently we can look forward to the future of manmade fibers with confidence and enthusiasm.

Report of the Tractor and

THE SAE Tractor and Implement Safety Light Subcommittee was formed in January, 1950, to work with the Farm Equipment Institute Committee on the lighting problems for farm machinery.

There are no reliable statistics on the number of farm-equipment accidents caused by lack of proper lighting on the vehicle, but it is very obvious that nonlighted, slow-moving equipment on the highway during the hours of darkness is a real hazard. Our problem was to work out those items which should be standardized to make a safe lighting system interchangeable among different machines.

What Lighting Is Needed

Considerable discussion was held as to what lighting was needed for protection and what part of the program should be standardized. In looking

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PROPER lighting of slow-moving farm vehicles—a necessity when these vehicles travel the highways at night—was the problem tackled by the SAE Tractor and Implement Safety Light Subcommittee and a corresponding FEI committee.

Among the results of their work, which is reported in this paper, are:

1. A standard breakaway connector for the tractor and implement lighting system.

2. A telescoping staff lamp that can be extended to the left, when needed for safety, to show the left-hand end of an implement.

3. Changes in the Uniform Act Regulating Traffic on Highways to permit use of such a staff lamp.

4. A lamp with an extension cord for use where a staff lamp is not long enough.

5. A proposed standard bracket for holding the extension-cord lamp.

The Author

B. G. VAN ZEE (M '38) is chief engineer of the Automotive Division of the Minneapolis-Moline Co. In 1922 he graduated from Iowa State College. In 1930 he received his master's degree in agricultural engineering from the same school. That year he joined the engineering department of Minneapolis-Moline. In 1936 he became assistant chief engineer and in 1944 chief engineer in charge of design on all tractors and industrial engines. He is a member of the SAE Technical Board.

Implement Safety Lighting Program

B. G. Van Zee, Minneapolis-Moline Co.

This paper was presented at the SAE National Tractor Meeting, Milwaukee, Sept. 16, 1953.

over the uniform traffic code at that time, we found that farm machines came under the following code:

"All vehicles not equipped with lights must be equipped with at least one lighting lamp or lantern exhibiting a white light visible from a distance of 500 ft to the front of such vehicle and with a lamp or lantern exhibiting a red light visible from a distance of 500 ft to the rear."

Code in Need of Revision

This code appeared to us to be in need of revision as it does not require any clearance lamps or reflectors, and as several states were considering doing something about a more positive code for farm equipment operated on the public highways during the hours of darkness, our work was lined up for us.

Distinctive Lamps Needed

The committee felt that we should have some type of special or distinct warning lamp or lamps for our relatively slow-moving machines on the highway as compared to automobiles. A flashing light was suggested as being a light giving the most protection to all. This was taken up with the National Committee on Uniform Traffic Laws and Ordinances, but it was found that it would be some time before approval could be obtained for farm equipment as the flashing lamp was reserved for school buses, fire equipment, and a few other emergency vehicles. Therefore, we decided to proceed without the special flashing lamp that we had hoped for.

From our discussions it was apparent that our first job was to develop a standard connector for the tractor lighting system so that everyone would have a common connector available. After considerable discussion of what the features for such a connector should be, the following specifications were written.

1. The connector must be double contact and polarized, with connection position easily recognized.
2. The connector should pull apart on a 2- to 6-lb pull, and wires must be sufficiently supported to resist this pull.
3. Connector must be corrosion resistant.
4. Service must be unaffected by dust and moisture, both when connected and unconnected.
5. Connector must be designed to permit one half to be mounted on the tractor.
6. It must be easily repaired.
7. Plugs and connectors adopted should be made on a free license principle.

The above specifications were sent to several manufacturers of connectors, and they were asked to submit a drawing of their proposal for a connector meeting these specifications. There were a couple of drawings submitted to the committee, and then a meeting was held with the connector manufacturers. After several meetings, we arrived at a drawing with tolerances that the various manufacturers said they could make, and we proceeded to ask for samples. The Cole-Hersee Co. of Boston was the only one that proceeded to build samples for test.

After a year of testing and improving this unit, during which time we discovered that the 2- to 6-lb pull was inadequate and had to be increased to 20 lb and that our polarized feature was not positive enough and had to be improved, this final unit was

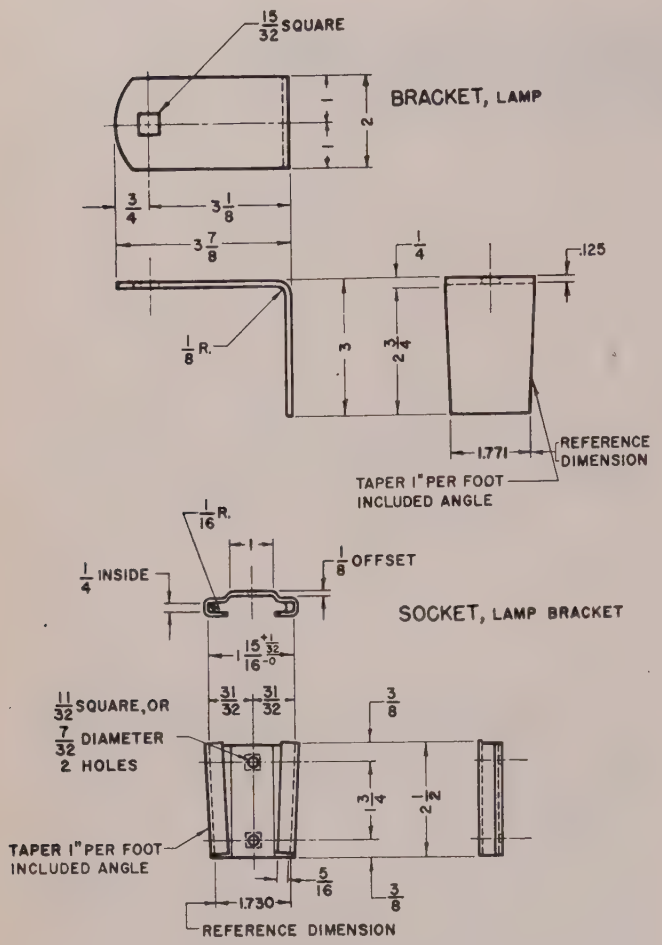


Fig. 1 – Proposed standard bracket for holding extension-cord lamp

then approved by the subcommittee on tractor lighting and presented to the SAE for adoption as a standard. This connector is now the SAE Standard Breakaway Connector and appears in the 1953 SAE Handbook.

Visibility Requirements

The development of a standard connector proved to be one of the easier parts of our problem in providing safe lighting of tractor implements on the highway at night. The requirement that the light be visible 500 ft to the front and 500 ft to the rear, and that the general outline of the drawn implement, especially the left-hand side, be visible to oncoming motorists was not so easy to solve. The mounting of a lamp or lamps on the rear of all drawn implements would become very costly, and as many implements would never be drawn on the highway at night, it was proposed to make this lamp or lamps an attachment which could be carried with the tractor and so be available to the operator at all times.

Extension Wire

The problem of what to do with all the extension wire when short-coupled machines were used brought out the suggestion that a staff lamp be used on the tractor, which could be extended out to a length of 70 in. and telescoped down to 35 in. in a transporting position when not being required to give protection to the drawn machine. This lamp would be mounted on the tractor as high as possible and then, when needed for safety, be extended out to the left to show the left-hand end of the implement. This looked like a step in the right direction, and units were made up for field test. It was realized that this type of lighting did not meet with approval in the present motor-vehicle law, so we decided to ask for approval of this type of a lamp for farm equipment from the National Committee on Uniform Traffic Laws and Ordinances.

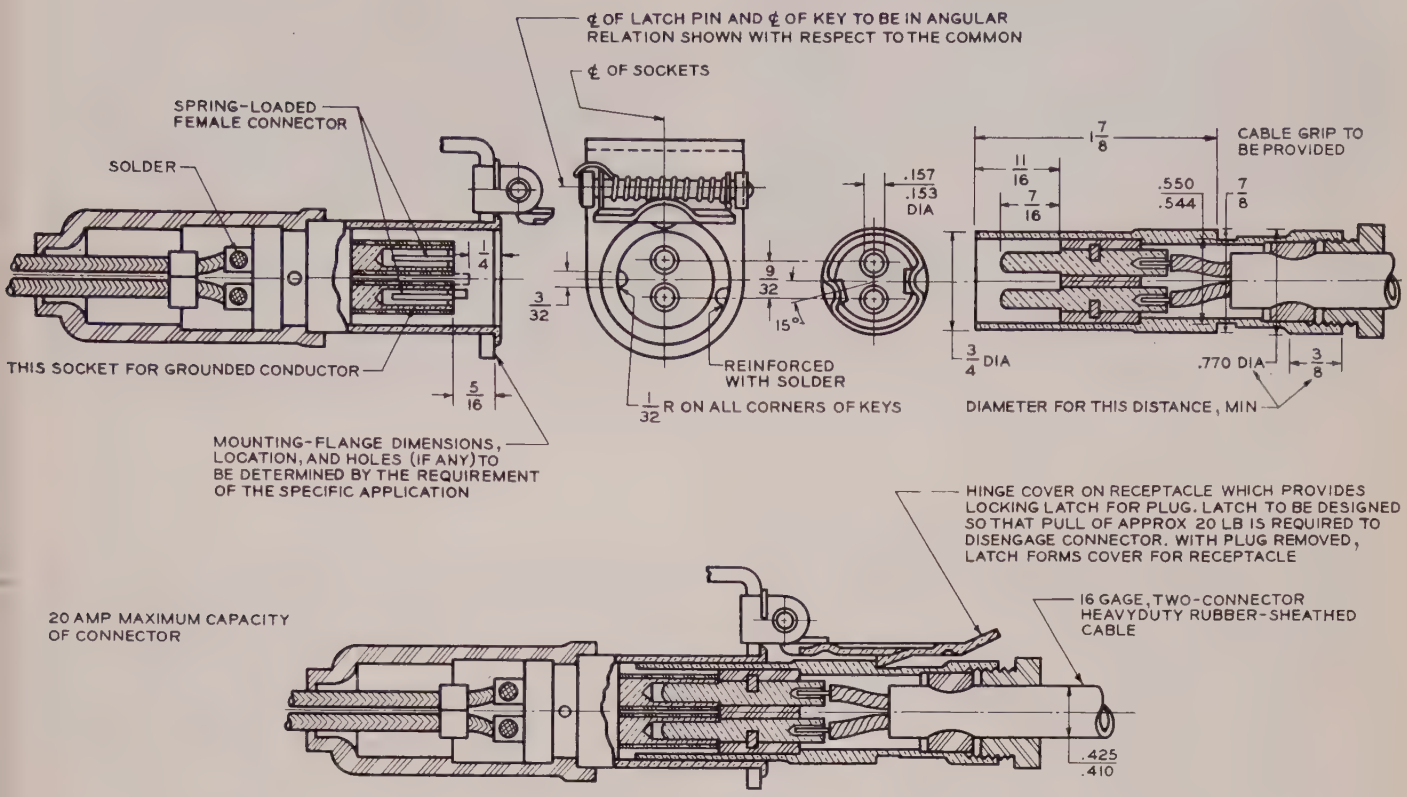
After several meetings of the Subcommittee on Lighting of the National Committee on Uniform Traffic Laws and Ordinances, (of which Martin Ronning, chairman of the FEI Lighting Committee, and the writer are members), a change and an addition to section 137 of the uniform acts regulating traffic on highways was agreed upon by the subcommittee. These changes were approved by the general committee and entered in Act 5, Uniform Act Regulating Traffic on Highways, where section 137 was amended and added to as follows.

In paragraph (a), the word in parentheses was deleted, and the words italicized were added. Paragraphs (b) and (c) are completely new additions.

Section 137. Lamps on other vehicles and equipment:

(a) All vehicles, including animal-drawn vehicles and including those referred to in section 123 (c) not (hereinbefore) specifically required *by the provisions of this article* to be equipped with lamps, shall at all times specified in section 124 hereof be equipped with at least one lighted lamp or lantern exhibiting a white light visible from a distance of 500 ft to the front of such vehicle and with a lamp or lantern exhibiting a red light visible from a distance of 500 ft to the rear.

(b) Every farm tractor not equipped with an electric lighting system shall at all times mentioned in section 124 be equipped with lamps or lanterns meeting the requirements of paragraph (a) above. Every farm tractor equipped with an electric lighting system shall at all times mentioned in section 124 display a red tail lamp and


℄ OF LATCH PIN AND ℄ OF KEY TO BE IN ANGULAR RELATION SHOWN WITH RESPECT TO THE COMMON
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SPRING-LOADED FEMALE CONNECTOR
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REINFORCED WITH SOLDER
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HINGE COVER ON RECEPTACLE WHICH PROVIDES LOCKING LATCH FOR PLUG. LATCH TO BE DESIGNED SO THAT PULL OF APPROX 20 LB IS REQUIRED TO DISENGAGE CONNECTOR. WITH PLUG REMOVED, LATCH FORMS COVER FOR RECEPTACLE
20 AMP MAXIMUM CAPACITY OF CONNECTOR
16 GAGE, TWO-CONNECTOR HEAVY DUTY RUBBER-SHEATHED CABLE
.425 .410

Fig. 2 – Breakaway connector

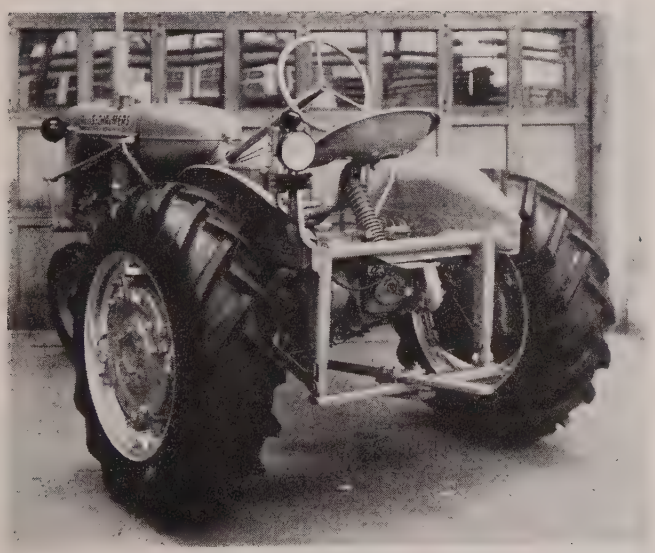

Fig. 3 – Allis-Chalmers proposed 3-way (warning and service) lamp with spool extension

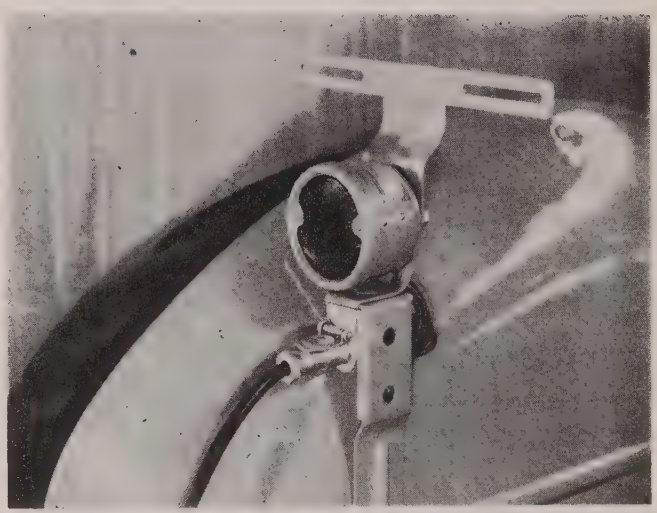

Fig. 4 – Breakaway connector mounted on Harry Ferguson tractor

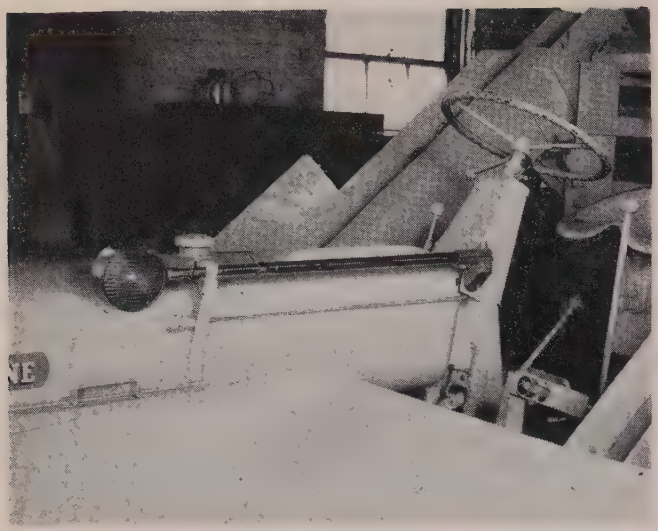

Fig. 5 – Proposed staff lamp on Minneapolis-Moline Model ZB tractor

Fig. 6 – Proposed staff lamp on tractor and reflector on Minneapolis-Moline pull-behind combine

either multiple-beam or single-beam headlamps meeting the requirements of sections 126, 141, and 143, respectively.

(c) All combinations of tractors and towed farm equipment shall, in addition to the lighting equipment required by paragraph (b) above, be equipped with a lamp or lamps displaying a white or amber light visible from a distance of 500 ft to the front and a red light visible from a distance of 500 ft to the rear, and said lamp or lamps shall be installed or capable of being positioned so that visibility from the rear is not obstructed by the towed equipment and so as to indicate the furthest projection of said towed equipment on the side of the road used by other vehicles in passing such combinations. And further, all such towed farm equipment shall be equipped either with two tail lamps displaying a red light visible from a distance of 500 ft to the rear or two red reflectors visible from a distance of 50 to 500 ft to the rear when illuminated by the upper beam of headlamps, and the location of such lamps or reflectors shall be such as to indicate as nearly as practicable the extreme left and right rear projections of said towed equipment on the highway.

With the above changes and additions in the traffic code we thought we were making good progress. During our 1952 summer tests of the staff lamp by various companies it was found that on certain machines, like the left-hand cut combines that extended to the left a considerable distance from the centerline of the tractor, this staff lamp did not fulfill the requirements of section 137 of the code.

From the reports of the various members of the committee, it became apparent that a lamp with an extension cord would be required to protect certain machines. Several ways were suggested and worked out to handle the extension cord and lamp between the tractor and the drawn implement.

Fig. 7 – Oliver 88 with safety lamp mounted on left wheel guard

Two Types of Lighting Demonstrated

In order to have a demonstration of the two types of lighting, staff lamp and extension-cord lamp, a meeting was held at the Hinsdale Farm of International Harvester Co. Various makes of tractors pulling different implements equipped with a staff lamp or an extension-cord lamp were viewed from an automobile driver's position as to their ability to pass as safe night-lighted vehicles. Reflectors were also used in conjunction with lamps on these vehicles to outline the extremity of the machines. This demonstration was very worth

Fig. 8 – Connector mounted on rear of Oliver 88 tractor

Fig. 9 – Breakaway connector mounted on Ford tractor

while, and good representation from the various companies turned out for this meeting.

In the discussion after the demonstration it was decided to leave the use of the various types of lamps, staff or extension cord, up to the individual manufacturer, but the standard breakaway connector and the standard bracket for holding the extension-cord lamp on the tractor be furnished by the tractor people either as standard equipment or as an accessory bundle. A drawing of the proposed standard bracket is shown in Fig. 1. This bracket will be attached by the implement people on the machine which requires extension-cord-type lighting for full protection and also on the tractor so that the lamp can be carried on the machine at all times. A drawing of the breakaway connector is shown in Fig. 2, and the socket part, which is mounted on the tractor, should be located within the following dimensions:

1. Height above the ground is to be 20 to 50 in.
2. Forward location from hole in standard SAE-ASAE drawbar for power take-off is to be 12 to 24 in.
3. Lateral location is within 14 in. of tractor centerline.
4. Voltage of the electrical system should be indicated on the tractor at the connector.

Many of the tractor companies are now supplying the socket part of the standard breakaway connector on various models of their tractors, and the bracket for the lamp should be coming very soon.

Some examples of the mounting of the breakaway connector and safety lamp are illustrated in Figs. 3-10.

Fig. 3 is the Allis-Chalmers proposed 3-way (warning and service) lamp with spool extension mounted on their "CA" tractor.

Fig. 4 shows the breakaway connector mounted on the Harry Ferguson tractor.

Fig. 5 shows a proposed staff lamp on Minneapolis-Moline model ZB tractor.

Fig. 6 shows the proposed staff lamp on the tractor and reflector on the Minneapolis-Moline pull-behind combine.

Figs. 7 and 8 show the Oliver 88 with the safety lamp mounted on the left wheel guard and the connector on the rear of the tractor.

Fig. 9 shows the breakaway connector mounted on the Ford tractor.

Fig. 10 shows a proposed staff lamp mounted on a Ford tractor.

This completes our report; and I want to thank the tractor and implement companies, also the lamp and connector people, for their fine cooperation in getting this job done.

Fig. 10 – Proposed staff lamp mounted on Ford tractor

Does the Public Benefit from

EQUITABLE size and weight legislation is that which fosters the most efficient use of the combination of the vehicle and the highway, to the end that the public will receive the maximum return on its investment in highways and bridges through convenience, celerity, and economy of transportation.

It is therefore most appropriate that we, as engineers, approach a consideration of the effects of vehicular size and weight regulations from the standpoint of the people at large, rather than from our own narrow viewpoint. The test of any type of legislation is whether it benefits the public.

If this is the issue, then the answer must be definitely in the negative. Beneficial results can hardly be expected from legislation premised on a fallacy, and present size and weight restrictions are lacking in any sound foundation, either scientific or economic. They are not consistent, and they give palpable evidence on their very face that the legislators who framed them had little or no conception of the essential economics of highway construction.

The author realizes that in thus voicing the conclusions which a study of this topic has compelled, he is departing from the form customary in engineering reports, which is first to trace the historical background of the subject and the circumstances which led to the immediate project, then to describe

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THIS paper describes how present size and weight regulations are based on capacities of inadequate highways, or on various arbitrary bridge formulas, or somtimes both. Within a state, contradictory weight limits often are prescribed; and uniformity from state to state is noticeably lacking.

Adapting the demand for bigger trucks and trailers for hauling more payload to the various size and weight limitations has resulted in some improvements in truck design. But what are needed are consistent regulations based on logic and fact to encourage designers to produce safer and more efficient vehicles – thus promoting the public interest.

As the regulations stand now, Mr. Horine says, they do not benefit the public.

The Author

MERRILL C. HORINE (M '17) is consulting engineer for the Mack Mfg. Co. Before joining Mack in 1918, he was employed by A. C. Stewart in 1907 and Stromberg Motor Devices Co. in 1910. He became assistant technical editor of *Motor Age* in 1912 and editor of *The Commercial Vehicle* in 1913. Mr. Horine served as a 2nd Lieutenant in World War I. His work at Mack for many years has been principally the education of salesmen in the engineering principles, manufacturing practices, and operating economics of the products of his company.

Size and Weight Restrictions?

Merrill C. Horine, Mack Mfg. Corp

This paper was presented at the SAE Summer Meeting, Atlantic City, N. J., June 9, 1953.

the means and methods employed in its pursuit; to trace the findings in detail and then, in conclusion, (if the original purpose has not by then been entirely obscured by this mass of detail) venture such conclusions as the study warrants.

In a written report, of course, the experienced analyst can always read the end of the paper first and thus gain an idea of what the writer's purpose is and what significance may be attached to the items of evidence or logic presented. At the outset of this paper, however, let there be no uncertainty concerning where this author stands:

1. Ultimate public benefit must be the gage of the soundness of any legislation or regulation.

2. Transportation is the prime essential of commerce and industry, and the whole progress of modern society toward a higher standard of living is dependent upon greater efficiency in production and distribution. Therefore anything which impedes or increases the cost of transportation harms the public.

3. Highways are of no value except as they serve transportation. In fact, they can properly be regarded only as an integral part of our transportation facilities. No part of a thing can have merit save as it functions in relation to the whole.

4. Legislation or regulation which seeks to place arbitrary limits on the efficiency of one component of our transportation system to preserve other and inadequate components and to retard their modernization serves in the long run to impede transportation and to increase its cost. It is therefore not in the public interest. Present limitations on the size and weight of highway vehicles do just this.

Regardless of special taxes levied on truck operators, it is the public in the end which pays for highway transport operation as well as the roads over which the vehicles run. Therefore, the more economically trucking service can be conducted, the less the public has to pay for everything it buys and the more profit potential there is in everything it produces and sells.

Large Loads Mean Low Unit Cost

There is no question that the cost per ton-mile of moving a load is inversely proportional to the amount of load carried. Two 5-ton loads will cost more to move a given distance in a given time than one 10-ton load. This is as true on the highways as it is on the rails, on the sea, or in the air. It is therefore in the public interest that highway vehicles of the largest practicable capacity for which loads can consistently be found be generally employed. But economical operation does not depend on the vehicle alone. The highway is an essential part of the economic equation.

The Optimum

Doubtless there is a point at which further increase in vehicle capacity would cease to be economical, and also it is to be expected that there is a limit to the highway load capacity which it would be economical to construct. Somewhere along these respective curves, as shown in Fig. 1, there will be a crossing. That is the point where the maximum economy of the road and the vehicles on it, considered together as an economic unit, will provide the lowest overall cost per ton-mile to move the public's goods.

Whether consciously or not, it is this optimum which, in their motor vehicle size and weight laws, our various states seek to attain. The question before us here is how successful these laws have been in this direction. Unfortunately, though numerous research projects have been started which seek to establish factual bases for evaluation of the

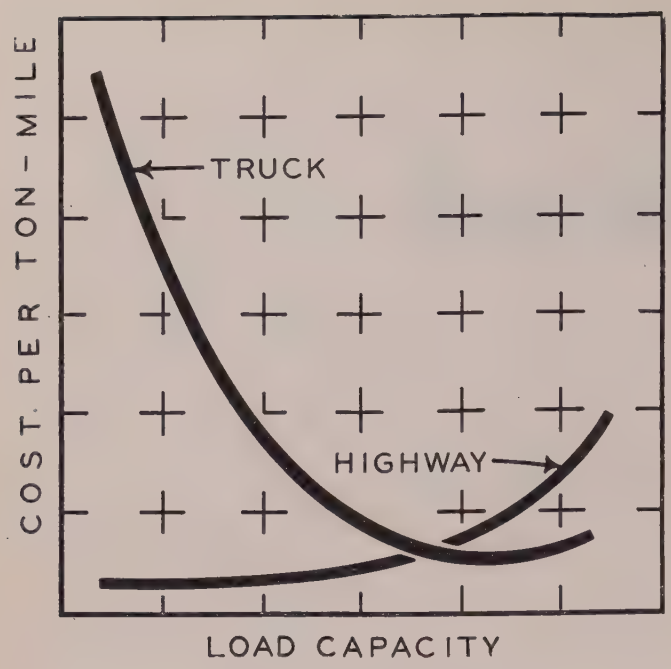


Fig. 1 – Cost per ton-mile versus load capacty

economic relationships between highways and vehicles, these have not as yet yielded comprehensive or conclusive results.

Perhaps the prime reason why present size and weight restrictions have failed to benefit the public is that they seem to have as their objects, first, the preservation of a highway system which, through shortsighted planning, political promotion, and 10 years of neglect during and following World War II, is inadequate; second, the singling out of a scapegoat – the heavy truck – for the failures of the politicians; and third, handicapping motor transport for the benefit of the railroads.

It is as though the wattage of electric current consumed by users were to be limited for the sake of preservation of transmission lines of inadequate capacity; the size and number of bathtubs restricted to conform with the capacity of superannuated water mains.

The public certainly will not benefit if, through underconstruction and undermaintenance, our roads and bridges are permitted to go to pieces any more than if, in the mistaken effort to preserve them, trucks are so restricted that the nation's transportation bill is increased far beyond what a sound and realistic highway program would cost. The salvation of our declining highway system is not in hobbling highway hauling, but rather in building and maintaining roads adequate for the economic demands of modern transportation – PAR, if you will.

Good roads pay for themselves. Better proof of this can hardly be found than in the record of the wholly self-sustaining Pennsylvania Turnpike, whose chief source of revenue is the heavy trucks, whose operators find that the economies realized from its use more than justify the tolls which they pay. This is in spite of the fact that they also pay heavy special taxes for the support of public roads which, in many cases, are inadequate for their use.

Where would marine transportation be if years ago the U.S. Engineer Department had set a limit of 14 ft of draft, 30 ft of beam, and 200 ft of length for all vessels entering our harbors to avoid the necessity of dredging wider and deeper channels?

Where would railroad transportation be if, for the preservation of trackage and bridge structures existing at the end of the Civil War, wheel loads had been restricted to the capacity of 40-lb rails, locomotive weights to 100,000 lb, speed to 35 mph?

How far could aviation get if airport runways were restricted to 1500 ft, wing loading per square foot to what it was in 1918, wheel loads to what could be safely supported on a cow pasture?

Demands on the Highway

Larger loads mean larger and heavier vehicles, and these make certain demands on the highways and bridges with respect to ability to sustain the resultant load concentrations and lane widths and curvatures to accommodate them. Were passenger cars and light commercial cars the sole users of the highways, roads and bridges adequate to provide the traffic space, freedom of flow, and strength to resist the destructive effects of the elements would be required.

No less an authority than Thomas H. MacDonald, late chief of the Bureau of Public Roads, testifying before the United States Senate some years ago, stated that roads constructed to withstand the elements successfully and adequate to sustain passenger-car traffic were capable of sustaining truck traffic with wheel loads of 9000 lb. He did not state how much greater wheel loads might successfully be sustained by roads of such type, properly built and maintained; nor has anyone to this date been able to fix such a figure.

We have been told, nevertheless, that axle loading beyond 18,000 lb will break down many existing pavements and that to build roads of greater weight capacity will increase their cost. Yet throughout those regions where greater axle loading is permitted, we have roads of the minimum standard which have been successfully carrying legal axle loads of from 19,000 to 22,400 lb for a great many years. We have also seen many instances of roads which have gone to pieces although never traversed by any but passenger vehicles.

Even the much touted Maryland road test failed to support the contention that 18,000 lb was the safe limit. Subjected to the equivalent of 10 years of normal service in a few months, and with axle loads far beyond the 18,000-lb figure, this stretch of highway, already in use for many years, is still in

service and still capable of carrying the traffic which flows over it.

True, it developed cracks where it had been improperly constructed, so far as the subbase is concerned; but these cracks have not destroyed or even seriously impaired road serviceability.

The amazing thing is that they were no worse considering that, contrary to normal procedure, this stretch of road was accorded no maintenance whatever during the test, and no proper provision for drainage of the unstable subsoil areas was made. At the turn-arounds, where the traffic was twice as great as on any other portions of the route, there was no damage.

So, in spite of specious assertions to the contrary, the Maryland road test, instead of validating the 18,000-lb axle limit, actually invalidated it. If we looked into the history of this peculiar fetish, we would find that its origins rest on still less substantial evidence consisting principally of the Bates road test in Illinois, made 30 years ago, with solid-tired army trucks; the impact tests conducted by A. T. Goldbeck for the Bureau of Public Roads at Arlington; and a theory expounded by Older in 1924, concerning the endurance of concrete beams under repeated stress. As though a pavement, resting solidly on its subgrade, were a beam!

When we get all through examining the flimsy arguments for the arbitrary 18,000-lb axle-load limit, we are met with the further generality that to build roads capable of sustaining greater weight concentrations will involve higher costs and that this higher cost will be for the benefit of a minority of all of the trucks operating.

Well, let's look at this matter of cost. In 12 eastern states, axle loads of 20,000 lb or more are permitted, and have been for many years. Have their highway costs been out of proportion with those of the other states?

According to all we have heard, any additional cost to sustain greater axle loads would certainly be confined to that of the pavement itself. Such heavy costs as right-of-way, grading, compaction, preparation of the subbase, center-striping, guard rails, signposting, drainage, and many others would not be affected at all. Larger vehicles do not require greater widths or easier turns, for at the prevailing volume and speed of passenger-car traffic, safety demands more lane width and curve easing than truck traffic has any need for. As a matter of fact, the larger the capacity of the average truck the fewer trucks are required to move a given volume of freight and so the less they require in the way of traffic room.

So, let's look at the second premise, that any extra cost involved in building roads of capacity adequate to sustain economical highway transport would benefit only the owners of the minority of trucks having the heavier axle loads. This is the greatest fallacy of all; and in advancing it, the advocates of this and other restrictions upon the economic development of highway transport reveal their narrow viewpoint and apparent inability to recognize that highway transportation benefits commerce and industry generally and all of the people, not just those who own and operate them for public benefit.

The fact is that, although in number the heavier trucks are a relative minority, in transportation service these big trucks make a major contribution to the broad transportation picture. They are the ones which haul the people's goods at the lowest unit cost. They are the ones which render an indispensable service in the development of our natural resources, such as in coal and metal mining, logging, oil field work, heavy construction and excavating, and the vast over-the-road movement of livestock, farm produce, milk, and general freight so indispensable to our economic life.

Trucks haul more tonnage than all other forms of transportation combined!

Those who operate our heavy trucks are in one of the most highly competitive fields of industry. They stand or fall on exceedingly narrow margins in their operating ratios. Competition forces the survivors to be efficient and economical if there is to be any profit from operations, and the chief gainers from this efficiency are the people at large whose goods they haul. So the issue is whether this vital economy of transportation cost is to be promoted and increased or discouraged and lessened.

Axle-weight limitations are not the only ones affecting the gross weight of vehicles, however. Wholly without justification, 30 states restrict axle weights below 18,000 lb when they happen to be placed adjacent to companion axles, as in truck bogies and trailer tandems.

Years ago, the Bureau of Public Roads conducted some tests in which it was determined that where axles were spaced closer than 40 in. apart, the sum of their weights imposed stresses in the pavement greater than either axle would by itself. Conversely, of course, these tests also established the fact that if the axles were spaced 40 in. or more, the pavement suffered no more stress than from a single axle. In other words, such a bogie, loaded to 18,000 lb per axle or a total of 36,000 lb, exerted no more stress on the pavement than a single axle imposing 18,000-lb pavement loading. That's where the 40-in. figure, which is so prominent in many of the state regulations, comes from.

Tires capable of sustaining 18,000-lb axle loads, however, according to Tire & Rim Association standards are 11.00-20 in size or larger. Such tires have an overall diameter, when new, of approximately 42½ in. and require at least 6 in. of clearance between them, making 48½ in. the closest spacing practicable. For proper skid-chain clearance, indeed, 50 in. is the minimum; so that there is really no significance in the 40-in. figure so far as any regulations are concerned, and hence, so far as pavements are concerned, no justification whatever for any limitation on bogie or tandem loads other than individual axle-weight limits.

But there is another prevalent weight limitation

which deals with tandem-axle loading from a different standpoint. This is the bridge formula.

The Bridge Formula

"Mother, may I go out and swim?
Yes, my darling daughter;
Hang your clothes on a hickory limb,
But don't go near the water!"

This familiar admonition is no more contradictory than the dual bases of weight limitation now in force in the 48 more or less United States. Some states set limits on the maximum weight per axle imposed on the road and then add a contradictory maximum gross weight. Some prescribe axle-weight limitations for the protection of the road and a formula to govern the loading of bridges. Some allow more gross weight on a tractor-semi-trailer than on a six-wheeled truck with the same number of axles. Most of them restrict the weight on a two-axled bogie to less than double the weight allowed on a single axle.

The effects of the lack of uniformity are many. Operators who must cross state lines find that liberality of restrictions in one state avail them nothing, because in all interstate operations, their operations are limited by the regulations in effect in the one state in which they are most severe. This necessarily places interstate operators at a disadvantage in relation to intrastate operators in the more favorable states. It handicaps them artificially in their competitive relations with other forms of carriers; and, to the extent that it requires special types of vehicle to meet the restrictions, it increases the price and decreases the availability of the vehicles.

While amnesty to existing equipment in the form of so-called "grandfather" provisions generally accompany drastic changes in these limitations or requirements, this relief is not always given and rarely remains in effect for a sufficient time to permit proper amortization of the older equipment.

Many of the regulations are impractical for full enforcement, so that the conscientious operator who scrupulously observes them is penalized in relation to those who violate them. Most serious of all, of course, is the fact that anything which contributes to higher cost of transportation without compensatory benefit to the public at large works against the general welfare, reduces the level of prosperity, and contributes to inflation.

Table 1 – An Example of Gross Weight Limits in Wisconsin

Type of Vehicle	Axle Weights			By Axle Weight	Axle Spread	By Formula
	Front	Rear	Trailer			
4-wheel truck	7500	18,000	—	25,500	16	42,000
6-wheel truck	9000	32,000	—	41,000	21	47,000
4-wheel tractor						
2-wheel trailer	7500	18,000	18,000	43,500	40	71,000[a]
4-wheel trailer	7500	18,000	32,000	57,500	40	71,000[a]
6-wheel tractor						
2-wheel trailer	9000	32,000	18,000	59,000	40	71,000[a]
4-wheel trailer	9000	32,000	32,000	73,000	40	71,000[a]

[a] State limit, however, is 68,000.

Although functionally a bridge is an integral part of the highway, structurally it is different. The pavement on the highway is merely a wearing surface, like the carpet on the floor. The supporting structure is the earth itself, and the pavement is supported under its entire lower surface. A bridge is different. Here, the pavement is a slab supported by crossbeams and girders and at spaced intervals. Between the cross-beams there is no support, so that the slab is subjected to bending stresses.

Obviously, on the highway it makes no difference how the axles are spaced, provided they are not less than 40 in. apart, for the stress in the pavement and the underlying subbase can never exceed that of a single axle. On a bridge, however, we are concerned with the total weight imposed between supports of a span. Obviously the farther the axles are apart, the less the concentration of weight will be.

Some years ago, bridge engineers commenced to worry about these weight concentrations, and they devised a formula intended to limit such concentrations. This formula is: $C \times (L+40)$, in which C is an arbitrary number of pounds, based upon the class of bridge to be protected and L is the length, in feet, from the center of the first axle to the center of the last axle of the vehicle or any combination of vehicles. The constant C varies from 650 to 850, depending upon which state you are in. Two states make this little pastime more interesting by substituting different constants than the mysterious 40. Ohio uses 47½ and Wisconsin 26, with a C of 1000.

In 1948 John S. Worley[1] pointed out some significant facts concerning this formula, which clearly indicate how superfluous it is when considered from a practical standpoint. He said:

"The design of the standard commercial vehicle is such, as a rule, that no two single axles or groups of axles are closer than 12 ft. Therefore bridge spans of 24 ft and less can never have a gross load of more than that of a set of dual axles. Bridges with spans of 100 ft or more are largely designed for the dead load – the weight of the bridge structure. The stresses in these long spans are increased but little by additional live loads, due to the fact that the ratio of the live load to that of the dead load is very small. So we see that bridges with spans less than 30 ft and more than 100 ft in length are little affected by an increase in gross loads as long as the axle loads are not increased.

"The critical bridges in our highway system are found in the range of spans between 30 and 100 ft

[1] "Future Commercial Highway Transportation," by J. S. Worley. Presented at the SAE Summer Meeting, French Lick, Ind., June 8, 1948.

in length. Of these, the numbers are relatively small."

Prof. Worley also observed:

"Bridges for 2-lane highways are designed to carry two continuous processions of vehicles with maximum gross loads, spaced according to practical operating conditions. . . . A continuous procession of vehicles with maximum gross loads is seldom found on a bridge, and still extremely remote is the possibility of two such processions of maximum loads being on a bridge at the same time, *the normal loads for which bridges are designed.*

"Conservative ultimate strength of materials, modified by factors of safety ranging from 200 to 400% are used. Occasional overloads of 50% can be carried without causing undue overstress."

From this, it is obvious that from a practical standpoint there is no need whatever to superimpose the bridge formula on the axle-weight limitation. That so many states have chosen to do so indicates again the alarming failure of those responsible to recognize that the bridge and the highway are essentially one. The contention that bridges require weight limitations independent of those applying to the road itself is a clear admission that, in some cases, highways have been built in which the load capacities of their incidental bridges differ from those of the roads connecting them. Thus either one or the other is either deficient or overbuilt, revealing a waste of taxpayers' money.

A tacit admission that the bridge formula is too complicated for the average enforcement officer is found in the fact that 26 of the 34 states and the District of Columbia, which have enacted a bridge formula limitation, publish it in the form of a table to eliminate the necessity of calculation.

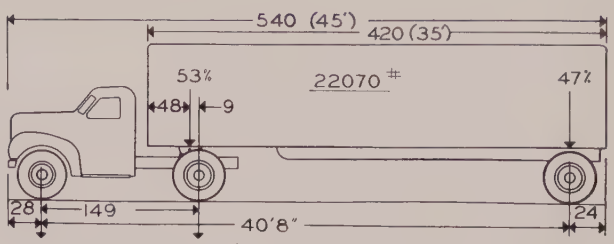
As though this dual basis of limitation were not enough, 43 states also have a definite ceiling on gross weight, regardless of the other two bases.

Just how contradictory these restrictions are can be visualized by considering how they apply to actual vehicle design.

For example, in the state of Wisconsin, the axle-weight limitation is 18,000 lb for single axles and 16,000 lb each for tandem axles. Irrespective of the weight on axles, however, a bridge formula, $(L + 26) \times 1000$, also applies, which results in the contradictions shown in Table 1.

With the single exception of the 6-wheel tractor, tandem-axled trailer combination, the bridge formula allows greater gross weight than the axle-weight limitation. It would therefore seem that the formula could be dispensed with in the interests of simplicity. However, there are those who point out that the bridge formula induces truck users to select types with the greatest possible axle spread; that if this inducement were removed, truckers would use shorter wheelbases and greater front and rear overhang, thereby bringing axle-weight concentrations closer together, to the detriment of the weaker of the bridges.

As a matter of fact, without the bridge formula

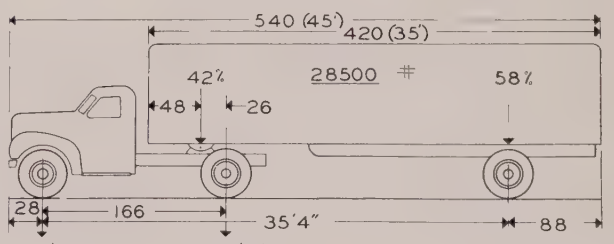


6%	94%			
9000	18000	CAPACITY	18000	45000
7000	7000	CHASSIS	2500	
700	11000	GROSS LOAD	10370	
7700	18000	G C W	12870	38570
		FORMULA LIMIT		64000

Fig. 2 – Normal 4-wheel tractor and single-axle semitrailer. Actual capacity is much less than capacities based on axle weight and bridge formula

it is certain that the extreme spread of axles which we see in many of the combinations now traversing our roads would never have appeared. To some extent, such a spread defeats its own purpose, for with the front and rearmost axles of a combination placed so far from the center of gravity of the load, a major percentage of the weight is thrown upon the intermediate axles. The definite limitations placed upon these, then, serve so to reduce the total gcw that a somewhat more judicious distribution of axles is desirable. While this would reduce the allowable weight under the bridge formula well below the impossible figures which it yields, it would permit the actual load to be increased.

A simple diagram will demonstrate the incompatibility often existing between the practical limits on gross weight as between the axle-weight basis and the bridge formula. In Fig. 2 is a fairly normal 4-wheel tractor and single-axle semitrailer, set up to provide the maximum formula allowance. The tractor has its front axle only 28 in. from the



15½%	84½%			
9000	18000	CAPACITY	18000	45000
7000	7000	CHASSIS	2500	
2000	11000	GROSS LOAD	15500	
9000	18000	G C W	18000	45000
		FORMULA LIMIT		60000

Fig. 3 – Tractor of Fig. 2 has trailer axle moved forward and greater fifth-wheel offset. Actual capacity now equals that on axle-weight basis

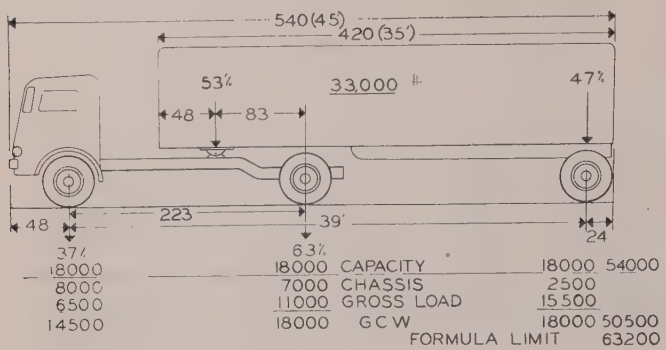


Fig. 4 – With 83-in. kingpin offset and trailer axle placed at rear, gross load is 33,000 lb

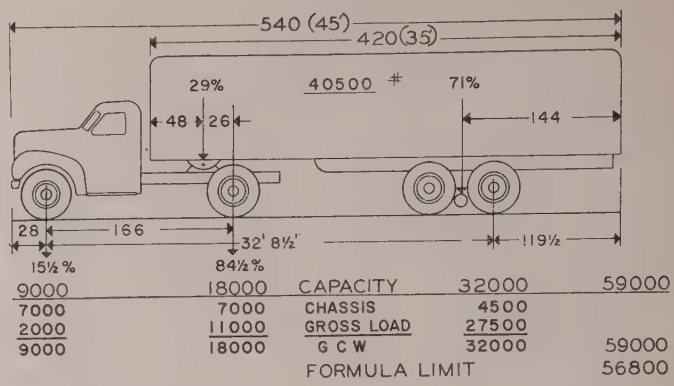


Fig. 5 – Four-wheel tractor and tandem-axle semitrailer designed to give maximum axle load greater than formula limit

front bumper, the trailer is a full 35 ft long, and the trailer axle is set 24 in. ahead of the extreme rear.

Although the tires and presumably the axles have a combined capacity of 45,000 lb which would be the limit by axle weight, the $(L + 40)$ x 800 formula allows 64,000 lb on the strength of the 40-ft plus axle spread. The actual capacity, without exceeding 18,000 lb on any axle, however, is but 38,570 lb. This is because the extreme rearward location of the trailer axle, in relation to the position of the fifth wheel – 48 in. from the front – throws 53% of the trailer weight on the fifth wheel. The conventional tractor, with a 9-in. kingpin offset, transfers 94% of this weight to the tractor rear axle, so that the weight on this axle is the limiting factor.

Assuming 7000 lb chassis road weight on the tractor rear axle, this leaves us 11,000 lb for gross load. As this represents 94% of the weight on the fifth wheel, this means that the front axle will carry 700 lb and the total weight on the fifth wheel will be 11,700 lb. Now since this 11,700 lb is 53% of the trailer gross load, the total will be 22,070, of which 47% or 10,370 will fall on the trailer axle.

Clearly what is needed here is more of the load on the trailer axle. This can be accomplished without reducing the load on the fifth wheel by moving the trailer axle forward. To be sure, this will reduce the axle spread and so the formula limit; but since 18,000 lb on the tractor rear axle is the limiting factor, there is no benefit to be gained from the 64,000-lb allowance by the formula.

Fig. 3 shows the same tractor with the trailer axle moved from 24 in. ahead of the tailboard to 88 in. and the fifth-wheel offset 26 in. instead of 9 in. This gives us 15½% on the front axle and 84½% on the tractor rear axle, for a fifth-wheel

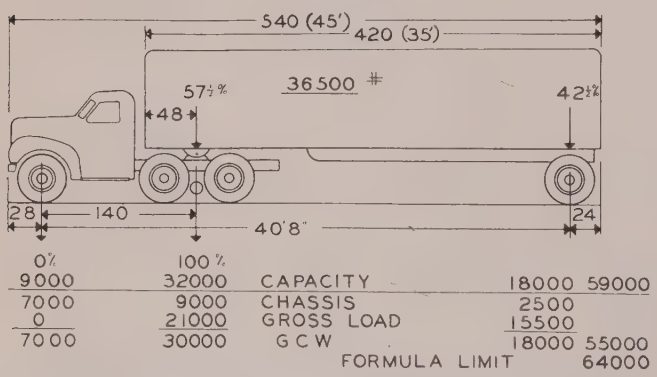


Fig. 6 – Combination of 6-wheel tractor and single-axle semitrailer affords greater formula limit, but full load on tractor bogie not provided by maximum loading

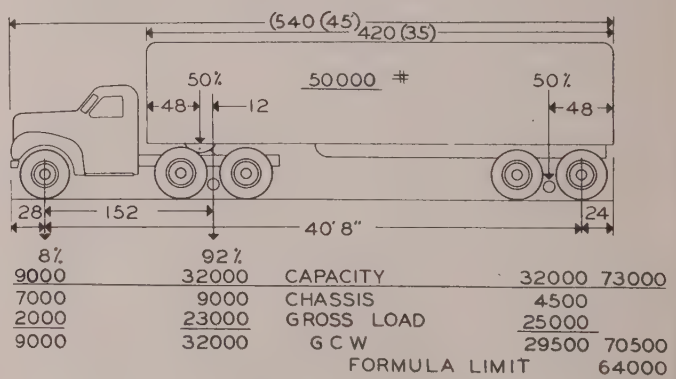


Fig. 7 – Maximum loading of this 6-wheel tractor and tandem-axle semitrailer combination exceeds formula limit

load of 13,000 lb. This, in turn, now represents 42% of the trailer gross load, for a total of 28,500 lb. Of this, 58% or 15,500 falls on the trailer axle, so that both the tractor rear and the trailer axles now carry their full allowable weight and the front axle its full capacity of 9000 lb, or 45,000 lb gross.

True, the axle spread has been reduced to 35 ft, 4 in., but the formula still allows 60,000 lb. Greater capacity than this without exceeding the axle limits would require more weight on the front axle. If we were to fit dual tires on the front wheels, which would be possible with power steering, we have 54,000-lb axle capacity. In Fig. 4 we have advanced the kingpin 83 in. ahead of the tractor axle and put the trailer axle back as far as possible, thus boosting the gross load to 33,000 lb and the gcw to 50,500. Our axle spread is 39 ft and the formula limit is 63,200 lb.

The situation is quite different when 6-wheeled tractors, tandem-axled trailers, or both, are used. Fig. 5 represents a 4-wheeled tractor with a tandem-axled semitrailer. It is possible to adjust the fifth-wheel offset and trailer tandem position so as actually to load the axles to the legal maximum, but in so doing the axle spread is a trifle shy of that required to qualify under the bridge formula.

Fig. 6 is an alternative in which an attempt is made to improve this picture by using a 6-wheeled tractor and a single-axled semitrailer. This provides the same axle capacity, of course, but since the trailer axle is the critical one, its maximum loading does not permit a full load on the tractor bogie, even with the kingpin directly over the trunnion, and the extra formula allowance produced by its greater axle spread is of no avail.

Fig. 7 shows what can be done with a 6-wheeled tractor and tandem-axled semitrailer. This combination affords 73,000-lb axle capacity, of which 70,500 can be utilized; but only 64,000 is permitted under the bridge formula.

Comparing Figs. 5 and 6, it is seen that the first, which, borrowing from locomotive parlance we may call the 1-1-2, has the advantage of 4000 lb more payload capacity. In favor of the 1-2-1, however, is the better traction in slippery going, the measurably greater safety, the better cab ride, and the lower price of additional semitrailers afforded by the 6-wheeled tractor.

In the third, or 1-2-2 type (Fig. 7) the payload is 9500 lb greater than the 1-1-2 and 13,500 greater than the 1-2-1, which will certainly justify the somewhat higher purchase price.

Table 2 – Size and Weight Limitations in State Motor-Vehicle Laws

	Length, ft			Weight, lb		Gross Weights			
Width, in.	Single Unit,	Tractor Semi-Trailer,	Other Combinations,	Per Axle,	Tandem Axles,	Gross Limit, lb		Bridge Formula	Table
96	35	45	45	18,000	28,000	42,000	Ky.	Ala.	Ariz.
Ala.	Ala.	Ala.	Ga.	Ala.	Minn.			Colo.	Ark.
Ark.	Ark.[a]	Conn.	Ill.	Ariz.		50,000	Conn.	Ga.	Calif.
Calif.	Calif.	Ga.	Maine	Ark.	28,650		Maine	Md.	Del.
Colo.	Colo.	Ill.	Minn.	Calif.	Miss.		N.H.	N.Y.	D.C.
Del.	Del.	Iowa	Miss.	Colo.[b]			Vt.	N.Dak.	Fla.
D.C.	D.C.	Ky.	Mo.	Fla.	32,000		Va.	Ohio	Idaho
Fla.	Fla.[a]	Maine	N.H.	Ga.[b]	Ala.			Wis.	Iowa
Ga.	Ga.	Mass.	N.Dak.	Idaho	Ariz.	52,600	Miss.		Kans.
Idaho	Idaho	Minn.	Tenn.	Ill.	Ark.				Maine
Ill.	Iowa	Miss.	Tex.	Ind.	Calif.	53,000	Mass.		Mass.
Ind.	Kans.[a]	Mo.	Va.	Iowa	Del.				Minn.
Iowa	Ky.	N.H.	W.Va.	Kans.[b]	Fla.	55,980	Tenn.		Mo.
Kans.	La.[a]	N.J.	Wis.	Ky.	Idaho				Mont.
Ky.	Mass.	N.Dak.		La.[b]	Ill.	56,000[d]	Ark.		Nebr.
La.	Mich.	Ohio	48	Mich.	Iowa				Nev.
Maine	Miss.[a]	Pa.	N.C.	Minn.	Kans.	58,400	Tex.		N.Mex.
Md.	Mo.	Tenn.		Miss.[b]	La.				Oreg.
Mass.	Mont.	Tex.	50	Mo.[b]	Maine	58,800	N.C.		S.C.
Mich.	Nebr.	Va.	D.C.	Mont.	Mich.				S.Dak.
Minn.	N.H.	W.Va.	Fla.	Nebr.	Mo.	60,000	Del.		Tenn.
Miss.	N.J.	Wis.	Ind.	Nev.	Mont.		Mo.		Tex.
Mo.	N.Y.		Kans.	N.Mex.	Nebr.		N.J.		Utah
Mont.	N.C.[a]	48	Mich.	N.C.[b]	Nev.		Okla.		Wash.
Nebr.	N.Dak.[a]	N.C.	Nebr.	N.Dak.	N.Mex.		Oreg.		W.Va.
Nev.	Ohio[a]		N.J.	Okla.	Ohio				Wyo.
N.H.	Okla.	50	N.Y.	Oreg.	Okla.	60,800	Iowa		
N.J.	Oreg.	Ark.	Okla.	S.C.[b]	S.C.				
N.Mex.	Pa.[a]	Del.	Oreg.	S.Dak.[b]	S.Dak.	62,000	Pa.		
N.Y.	S.C.[a]	D.C.	Pa.	Tenn.	Tex.				
N.C.	S.Dak.[a]	Fla.	R.I.	Tex.[b]	Wash.	63,800	Kans.		
N.Dak.	Tenn.	Ind.	S.C.	Utah	W.Va.		W.Va.		
Ohio	Tex.	Kans.	S.Dak.	Va.	Wis.				
Okla.	Va.	La.	Vt.	Wash.	Wyo.	64,600	Fla.		
Oreg.	Wash.	Mich.		W.Va.			Nebr.		
Pa.	W.Va.[a]	Nebr.	55	Wis.[c]	32,200		S.Dak.		
S.C.	Wis.	N.Y.	Md.	Wyo.	Oreg.				
S.Dak.		Okla.				65,000	Md.		
Tenn.	36	Oreg.	60	19,000	33,000				
Tex.	Ind.	R.I.	Ark.	Ohio	Utah	65,400	D.C.		
Utah		S.C.	Calif.						
Vt.	40	S.Dak.	Colo.	20,000	36,000	66,500	Minn.		
Va.	Ariz.	Vt.	Del.	Del.	Colo.				
Wash.	Minn.		La.	Pa.	Ga.	68,000	Wis.		
W.Va.	N.Mex.	55	Mont.		Ind.				
Wis.	R.I.	Md.	Ohio	22,000	Ky.	68,300	S.C.		
Wyo.	Wyo.		Utah	D.C.	N.C.				
		60	Wash.	Maine[b]	N.Dak.	68,000[d]	La.		
102	42	Calif.	Wyo.	N.H.	Tenn.				
Ariz.	Ill.	Colo.			Va.	72,000	Idaho		
Conn.		Idaho	65	22,400			Ill.		
R.I.	45	Mont.	Ariz.	Conn.	38,000		Ind.		
	Conn.	Utah	Idaho	Md.	D.C.		Wash.		
	Maine	Wash.	N.Mex.	Mass.					
	Utah	Wyo.		N.J.	40,000	73,200	Mont.		
			None	N.Y.	Pa.				
	50	65	Nev.	R.I.		73,900	Wyo.		
	Vt.	Ariz.			44,000				
		N.Mex.	Prohib-	None	N.H.	76,800	Ariz.		
	55		ited	Vt.			Calif.		
	Md.	None	Ala.		44,800		Nev.		
		Nev.	Conn.		Conn.		N.Mex.		
	None		Iowa		Md.				
	Nev.		Ky.		Mass.	78,000	Ohio		
			Mass.		N.J.				
					N.Y.	79,800	Utah		
					R.I.				
						80,000	R.I.		
						None	Ala.		
							Colo.		
							Ga.		
							Mich.		
							N.Y.		
							N.Dak.		

[a] Six-wheelers permitted greater length.
[b] On balloon tires only.
[c] On Class A highways only.
[d] Plus weight on front axle.

Are Uniform Restrictions Desirable?

If all states had identical limitations, each manufacturer could adapt his designs in this manner and provide users with the maximum capacity possible under the law. Such uniformity is very far from existing, and it is to be hoped that it will not come to pass in the foreseeable future. This is because the circumstances with respect to soils, terrestrial conformation, road and bridge development, and traffic conditions differ widely in various sections of our country. That there is some measure of consistency in the differences between the legislative provisions of groups of states is evident upon noting the regional pattern which is discernible.

In the northeastern states, where soils generally are rocky and well drained and where traffic conges-

tion is greatest both in the closely contiguous cities and towns and on intercity highways, axle-weight limitations are considerably more liberal, while length limitations are more restrictive.

In the wide-open regions of the Far West where, save for a few large cities, population and hence traffic is sparse, extreme length of combinations is not only permitted but encouraged for the sake of bridge structures which are often on the light side.

Thickly settled states of rolling terrain, where a multitude of obsolete bridges stand little chance of early modernization, seek limitations on both weight concentration and length. Table 2 shows how such a pattern can be traced.

No one seems to know just how much necessity there is for severe restrictions in those states whose highway and bridge systems are below par; but every informed person knows how seriously both traffic congestion and cost of highway transport would be affected, were the more liberal regulations in the northeastern states to be made more restrictive.

So much for weight restrictions, which are but one class of limitation affecting the character and efficiency of the highway vehicles which serve the public. There are other restrictions, such as height, which at the moment is not onerous; weight per inch of tire width, a carry-over from solid-tire days which has no present-day significance; number of units in a combination, which is of minor importance; and a host of other regulations not concerned with size and weight and therefore not germane to the subject of this paper.

Limitations on width and length, however, are worth some very close scrutiny. Only three states permit more than 96 in. of width, while the length of single units ranges from 35 ft in 36 states to as much as 55 ft in Maryland. Only 11 of the first group permit 6-wheelers greater length than 4-wheelers. Similarly tractor-semitrailer length limits vary from 45 ft in 21 states to 65 ft in Arizona and New Mexico. That of other combinations remains at 45 ft in 13 states and ranges up to 65 ft in Arizona, Idaho, and New Mexico. Five states prohibit other combinations.

Great benefit, both to the users of motor vehicles and to the public in general, would attend a greater measure of both uniformity and consistency between the regulations of states within regional groups, such as the Pacific Slope, the plains states and upper Mississippi Valley, the Gulf and southwestern region, and the industrial East and South.

In the meantime, it is necessary for the industry to deal realistically with the situation as it exists. To the truck engineer, the problem is how to design to provide the vehicle with the greatest productiveness in conformity with legal limitations on size, weight, and conformation – how to reconcile engineering balance with political restraints.

Two problems are involved: gross combination weight and overall length. In view of the predominance of 35 ft as the limit of length for a single vehicle and 45 ft as a combination, our universally adaptable chassis would have to come within these two limitations. In those states where greater unit lengths and greater combination lengths are permitted, just as in the states which permit up to 22,400 lb per axle, the basic design can be modified for operation within the areas where such liberality exists.

The principal problem is to design equipment which will permit operators in the majority of states to take full advantage of the possibilities contained in the weight laws.

Some Beneficial Effects

Of course all governmental regulations and restrictions are not necessarily evil. Many of the legal requirements and limitations are sound and beneficial – particularly those which promote safety of operation, prevent genuine abuse of highways and other public structures, protect the public health, facilitate free flow of traffic, restrict uneconomic "wild-cat" competition with legitimate carriers, and assure responsible service to shippers at fair rates.

In so far as these enactments and orders have spurred engineers to creative thinking and challenged their ingenuity, they have been of some general benefit to the profession. Out of their efforts to meet these challenges have come developments of advantage in themselves. Several examples might be cited.

Strict axle-weight limitations have spurred designers to develop means of increasing the proportion of weight on front axles, which have led to greater compactness of front-end construction, improvements in steering, greater fifth-wheel kingpin offset, and great improvements already made and in process in COE arrangements. These in turn have brought benefits of their own.

Increased front-axle loading has meant better brake balance, less tendency to skid, and greater certainty of steering. Greater kingpin offset has been found to provide better control in backing and interchange of trailers, reduced tendency to jackknife, and other advantages. Greater demand for COE construction has made it commercially expedient to develop improved accessibility, better vision, ventilation, and insulation.

Similarly, close enforcement of axle-weight limitations has resulted in a great increase in the use of tandem axles for trucks, tractors, and trailers. This rise in volume of production has implemented many advantageous improvements in their detail design, giving better balance of torque reactions, reducing the tendency to rear, stub, or hop; better brake balance and distribution of driving effort; improved traction; greater flexibility; and better alignment.

Even higher taxes have had beneficial reactions. Economic compulsion has accelerated improve-

ments in the fuel economy of engines, more scientific ranges and intervals of gear ratios in transmissions and driving axles. Doubtless, all of these things would have been brought about in response to the competitive urge of a free market. It may be, however, that such developments have been accelerated by the urgency of legislative and regulative compulsion, but it is certain that their natural evolution would have been along sounder lines than has sometimes been the actual case. Certainly, strict limitation of gross weight has spurred the efforts of designers to reduce the tare weights of chassis, of components, of trailers, and of bodies, since only in this way can the ultimate in payload capacity be realized.

Problems of Weight Reduction and Undue Shortening

Keeping tare weight to the minimum is desirable from the standpoint of economical operation, regardless of weight restrictions, for it obviously costs money to move weight, and the only weight that pays off is payload. For this reason truck designers naturally seek continually to eliminate excess or unnecessary weight. There are two ways in which weight may be saved:

1. By using lighter materials.
2. By using less material.

Lighter materials include aluminum, magnesium, and titanium; but their use is not so simple as merely substituting them for heavier kinds such as iron, steel, brass, bronze, copper, and zinc. The relationship between the various engineering properties of different materials varies considerably.

Aluminum of given tensile strength has much lower endurance, modulus of elasticity, and hardness than steel. Therefore, when aluminum is substituted for steel, for example, it is usually necessary to redesign the part completely, to use heavier sections than strength alone would require.

Stainless steel could be substituted for copper in radiator cores, with a saving in weight for equal capacity and convecting area, but it would be deficient in heat-transferring ability. So a stainless-steel core would have to be made bigger than a copper one for the same cooling ability, and this would probably make it heavier.

In some applications, certain materials are wholly impracticable. For high stresses, particularly where there are size limitations, for severe temperature conditions, and where dimensional stability is critical, the choice of materials is definitely limited to iron or steel and their alloys. The light metals so far exhibit little hope of adaptability to such parts as wristpins, connecting-rod bolts, valves, or valve lifters. Crankshafts, camshafts, and transmission gears require such physical properties quite beyond the scope of aluminum, magnesium, or titanium.

Weight is deliberately sought in some parts, such as flywheels, crankshaft counterweights, and torsional vibration dampers. Weight saving in these parts would result in destructive vibration, which would reduce the durability and reliability of associated parts. Brake drums of sufficient strength could be made of steel or a combination of steel and cast iron; but experience has shown that only highly specialized iron castings of considerable weight can provide the antiscoring, checking, and cracking qualities essential to withstand the high and rapid input of heat essential to the effective functioning of the brakes.

On the other hand, where stresses—both mechanical and thermal—are low, where high temperatures are not encountered, and where wear is not a factor, the light metals possess considerable advantage over the ferrous metals, such as reduced mass, stiffness without weight, good heat conduction, and resistance to corrosion.

Alternate materials often greatly complicate manufacture. Nonferrous metals lend themselves to die casting, extrusion, and spinning better than ferrous metals; but some are inferior in drawing ability, are limited in their response to heat-treatment, hardening, and tempering. They are difficult, and in some cases impossible, to solder and to weld and require difficult and more expensive processing in machining and forging. These difficulties, plus the greater cost of the material itself, usually make the use of light metals considerably more expensive than that of steel or iron.

Weight saving is possible without changing the nature of the materials, however, by more scientific design. The aircraft industry has accomplished wonders in this direction. Thin sheets of stainless steel can be made to perform the functions of heavier panels of conventional steel. High-strength steel alloys, coupled with greater care in design to eliminate stress concentrations and develop maximum section stiffness, can considerably lighten highly stressed parts. Box girders provide equal strength and superior rigidity to channel or I-beam sections, with less weight. Bored-out shafts, scientifically ribbed castings, and strengthening processes such as shotpeening, graduated heat-treatment, and ground surfaces preserve or enhance strength and endurance while saving weight.

The main difficulty with all of this is, though, that almost always such weight saving means increase in the cost of the parts. It is customary for common-carrier truckers to evaluate weight savings as worth from one dollar to $1.87 per year in extra revenue for every pound of tare weight saved. Accepting this standard, the industry has made a great deal of progress but has of necessity had to stop short when the extra cost per pound of weight saved necessitated an increase in price to the customers, which would offset the possible revenue gains.

There is no need of legislative compulsion to induce manufacturers to strive in this direction. Simple competition is the most powerful spur that can be brought to bear. But legislative compulsion

has inspired the truckers to demand more and more weight saving and to agree to some important sacrifices to obtain it.

This has impelled some producers of vehicles to go to extremes to bring down their chassis weights, some body and trailer builders to do likewise. In many cases, conservative margins of strength, endurance, and durability have been seriously reduced, thus impairing safety, reliability, and long-run economy.

Operators have sometimes accepted the resultant need for more frequent inspection, lubrication, adjustment and replacement, faster tire wear, greater difficulty of control, and decreased comfort and protection of drivers to avoid the necessity of reducing their payloads and thereby their revenues under more stringent weight limits. With more reasonable and equitable regulations, on the other hand, they would have been unwilling to do so and would have recognized that, in the present state of the art, reduction in weight beyond the economic limit ceases to produce *light-weight* vehicles but produces instead *short-weight* vehicles.

A few examples: A wide front tread provides the utmost in stability, permits a shorter turning radius, and affords slightly better accessibility to the powerplant; but a narrower tread means a shorter axle and therefore a lighter one.

Long, wide springs afford the easiest riding and materially reduce road impact; but they weigh considerably more than short, narrow ones with a relatively high pile, so driver comfort and protection of the load from shocks are impaired, and shock on the chassis, on the tires, and destructive pounding of the road are intensified by weight paring.

Twenty-two-in. wheels and tires roll more easily, decrease shocks, afford longer tire life and better brake cooling; but all of these advantages must be sacrificed for the sake of the lighter weight of the 20-in. size.

Too much shortening can spoil a cake, and it can also prove deleterious to a truck design. Length limitations of unwarranted severity have given rise to a quite reasonable insistence from users that as little of this precious length as possible be consumed with the nonrevenue space required for the cab and powerplant. Unfortunately this demand has coincided with insistence upon greater power, particularly in diesels, which usually are longer than gasoline engines of the same power.

Some worth-while development has taken place as a result of this, such as the present trend toward higher specific output from engines, thus producing more power from engines of the same length. The unnecessary ballooning at the front ends of hoods which passenger-car stylists foisted upon the truck industry has been discouraged and there is a healthy trend back to shallow, flat radiator shrouds or exposed radiators.

With a 45-ft overall length limitation, operators demand a 35-ft semitrailer. This leaves 10 ft from the front of the trailer to the front of the bumper, of which a minimum of 18 in. is required for cab clearance, leaving 102 in. maximum length from the front of the bumper to the back of the cab. Heretofore this dimension has been from 108 to 120 in., which means that from 6 to 18 in. had to be subtracted.

First of all, the waste space in the radiator cowling had to be eliminated, bringing the grille or shutter assembly up as close as possible to the radiator. Then the customary 4 to 7 in. from the grille to the front of the bumper had to be cut down. Cab lengths are difficult to reduce in view of the insistence upon driver comfort. Very little can be saved by flattening the fan shroud and bringing the radiator closer to the block. This leaves only one remaining place where length may be reduced and that is in the rear cylinder pocket in the dash or firewall.

According to the exigencies of their individual designs, different manufacturers who have responded to this 102-in. dictum have resorted in varying degrees to all of these means. The results have not been altogether happy in all cases.

The flattened appearance of the front end is distinctly noticeable in all cases, but has been accepted in good spirit by practical-minded operators. The bumpers, however, nearly flush with the grilles, have certainly lost a good deal of their protective value. In some cases the shallower fan shrouds must certainly have resulted in impaired cooling, and the problem of fan-belt replacement has surely been made more difficult. One of the greatest drawbacks has been the pocketing of the rear of the block inside the cowl. There are designs in which this has been accomplished without sacrifice, but in most cases it is impossible to rebore the rear cylinder with the engine in place.

The COE Tractor

Another approach to this problem has been to resort to the cab-over-engine arrangement, whereby the distance from front of bumper to back of cab is reduced to somewhere between 96 and 69 in. Owing to the restriction of unit lengths to 35 ft in most states, however, this reduction does not permit any increase in load space save in those few states where combinations are allowed more than 45 ft overall.

Some of the COE designs have the engine located beneath the seat and the door opening a considerable distance ahead of the front axle. This is desirable in city service and in states where axle weight rather than formula governs the gross weight. But in a growing list of states, in which the formula is the governing limitation, the setback front axle serves to reduce the total spread between axles and therefore the permissible gross weight.

Consequently, special COE types intended for

over-the-road tractor service only have been developed, in which the front axle is placed as far forward as possible, which renders them as poorly suited to city service and at as great a disadvantage in localities where axle weight is the ruling limitation as the previously mentioned type is handicapped in the formula states. Forward axle location also seems to have some advantage in that it favors the forward-swinging door, preferred by tractor drivers because of its convenience in backing and in that riding ease – never too good in short-wheelbase 4-wheeled COE chassis – is somewhat better than where the driver sits slightly ahead of the axle.

One thing which has retarded the adaptation of the COE to over-the-road tractor service has been the relative inaccessibility of the power plant for servicing, characteristic of the type heretofore. Various means have been employed to alleviate this condition, including power plants and cabs which slide out the front like dresser drawers; cabs that tilt forward, sidewise, and even backward; and cabs that lift straight up.

Various complications have been introduced in these designs, such as hoists and jacks to move the cabs or powerplants; intricate powerplant and control connections; swinging, telescoping, and demountable steering controls; elaborate hinged scuttles; folding seats; removable panels; swinging fenders; and the like. Some cabs have of necessity been made of sleeper length, and in all cases the cost and weight have exceeded those of the conventional type considerably.

Of course, where the tractor is of the 6-wheeled type, the difficulty about riding ease is overcome, owing to the inherent superiority of the 6-wheeler in this respect.

Some improvement in powerplant accessibility has been achieved by mounting the cab much higher than usual, which has found favor in itself with over-the-road drivers, and means are being developed whereby the cab is made quickly tiltable without the amount of complication formerly thought necessary. Some very large types have been developed particularly for service in the western states, in which the cab is of sufficient size that excellent accessibility is achieved without any tilting, lifting, or sliding of either the cab or the powerplant.

At best the COE saves less than 3 ft of length, and it is highly debatable whether it would have been resorted to for tractor work had the tractor-semitrailer length limits been uniformly 50 ft or more.

Need for Greater Width

Restrictions on overall width of vehicles reflect the greatest degree of uniformity in the entire body of state motor-vehicle laws, since with only three exceptions, 96 in. is prescribed as the limit. Arizona, Connecticut, and Rhode Island permit 102 in.

In 1941, SAE adopted a Standard on Load and Dimension Limitations on Motor Vehicles setting 96 in. as the maximum width of body; but 102 in. as the maximum across the tires (Fig. 8). This was in recognition of the fact that because of sidesway, the width at the top of the vehicle is more critical than that across the tires and because tires to accommodate axle loads of 22,400 lb, with proper center spacing between the duals and adequate spring widths and clearance in connection with the standard 34-in. frame, require 102-in. overall width. This standard has since been canceled for reasons having nothing to do with its merit.

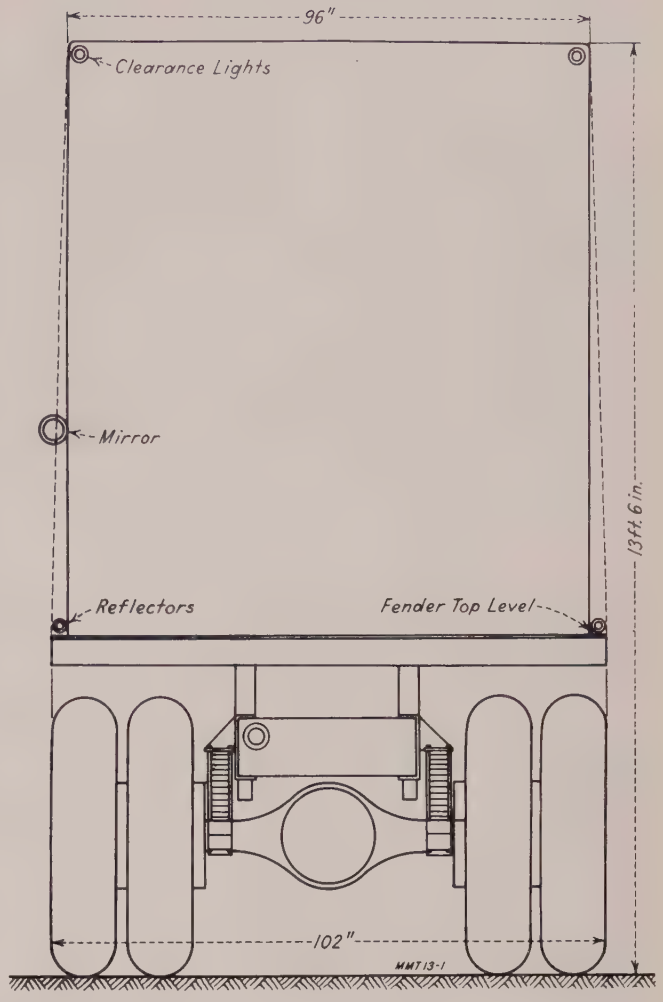


Fig. 8 – Maximum widths provided by SAE Standard on Load and Dimension Limitations on Motor Vehicles

Of course, the 22,400-lb axle limit applies only in Connecticut, Maryland, Massachusetts, New Jersey, New York, and Rhode Island. However, the District of Columbia, Maine, and New Hampshire allow 22,000 lb and Delaware and Pennsylvania, 20,000

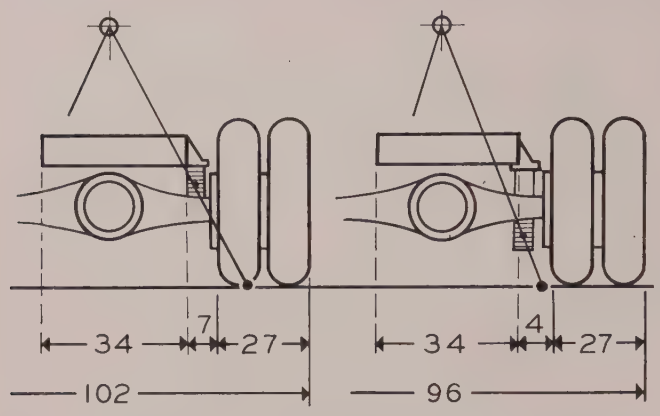


Fig. 9 – High spring position and maximum spring center distance provide maximum stability

lb. This makes 11 states in which tires of the 12.00 -in. cross-section may be required for full permitted axle load.

It might seem, nevertheless, that, since 36 states limit axle weight to 18,000 and one, Ohio, to 19,000 lb, for which the 11.00-in. cross-section is ample, there would be little need in those states for a more liberal limit on width. However, it should be borne in mind that in the past we have seen a trend toward greater cross-sectional tire width and lower inflation pressures in the interests of better cushioning, improved flotation and traction, increased economy, and greater safety.

At the moment, no further increase seems imminent, but in view of the fact that ground pressure per square inch of contact is a direct function of inflation pressure, it would be very much in the interest of the states as a highway conservation measure to foster the development of wider tires and lower inflation pressures, entirely aside from considerations of safety, conservation of vehicles, and protection to load and operating personnel.

It is possible to design vehicles to operate on 12.00-size tires within 96 in. overall width, either by employing narrower-than-standard frames or by mounting the springs beneath the side rails instead of alongside; but the penalty is reduction in stability. Maximum stability is achieved by using not only the greatest spring center distance that the space available permits but also by placing the springs as high as possible, as Fig. 9 illustrates.

Some enforcement officials have insisted upon measuring overall widths over the tire bulge at the ground, which is not only devoid of any practical justification but serves in effect to reduce the allowable overall width at hub height to as little as 94 in.

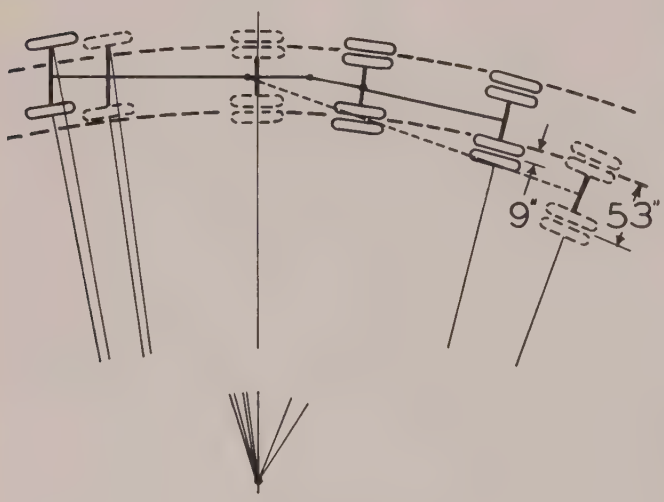


Fig. 10 – Comparison of off-tracking of truck-full-trailer and tractor-semitrailer combinations

Turning Radius and Off-Tracking

Among the most serious difficulties which the formula type of gross weight restriction has produced is the effect of exaggerated axle spread upon turning radius and off-tracking of tractor-semitrailer combinations.

Off-tracking is the deviation of the rearmost wheels of a vehicle from the path of the front wheels in rounding a turn. This deviation is always at the inside of the turn and increases with the distance between axles and the sharpness of the turn. Although the short wheelbase of the tractor provides it with a relatively small turning radius, that of a combination remains relatively large, since the off-tracking of the rearmost wheels so widens the total turning path of the combination that inner curb clearance limits the sharpness of the turn.

Generally, the shorter the distance between axles of any rigid vehicle, the smaller the turning radius will be and the less off-tracking will result. Unless we wish to steer the rear wheels of a combination, like those at the back of an aerial hook-and-ladder truck, there is not much we can do about this off-tracking.

Were the bridge formula restrictions to be lifted and the simple common-sense axle-weight basis alone relied upon, then we could reduce our axle spreads for the same overall lengths and in some cases reduce the latter as well and thereby decrease the amount of off-tracking.

It is an interesting fact in this connection that the truck-full-trailer combination of equal load platform length and axle spread off-tracks less than a tractor-semitrailer. Fig. 10 indicates this. Here we have the two types shown negotiating a turn at a 120-ft radius. Each type has 35 ft of load

platform length, the semitrailer in one unit and the truck-full-trailer in two units. The latter offtracks 53 in.; the former, 9 in.

Notwithstanding this, five states prohibit full trailers. This is unfortunate in another way, furthermore, since the two axles on a full trailer are permitted in most states to carry 36,000 lb compared with 32,000 on a semitrailer tandem.

Is This the Answer?

For a while, in the course of this study, it seemed that the solution of this whole problem might lie in an adaptable combination, such as shown in Fig. 11. With this, if the driver spied a group of inspectors or state troopers bearing tapemeasures, he could pull off the road long enough to crank the front axle forward, the trailer tandem back, advance the nose of the trailer, and reduce the kingpin offset, possibly shortening the body as well. On the other hand, if the boys are gathered around a platform scale or are lugging loadometers, then he would move the front axle back, increase the kingpin offset, reduce the fifth-wheel overhang, and push the trailer tandem forward.

Admittedly, such a design might involve a few engineering problems; but the project was abandoned because it was discovered that many of the highwaymen were so unethical as to ply both tapes and scales. The usefulness of one feature of the contrivance, however, remains to be investigated. That is the extensible trailer idea, considering the wide variation in allowable lengths in the different states.

And so we come back to the question which is the title of this paper: "Does the Public Benefit from Size and Weight Restrictions?" The answer is, that they do not. Another question might be, could they? Some restriction is doubtless necessary in the public interest. The restrictions we now have have given us some benefits in the way of stimulation of truck and trailer development, if nothing else. Greater consistency in those restrictions would mitigate some of the harm they now do. If simplified and based on logic and fact, many of their present defects would disappear.

In that happy event, designers would be encouraged to produce vehicles with more favorable and efficient load distribution, greater stability and maneuverability, and a higher degree of safety than present conflicting restrictions suggest. Greater uniformity of regulations would induce a greater uniformity of product with resulting gains in efficiency and economy. These things can be accomplished without impairing the economy or adequacy of our road systems.

Truck producers and operators do not ask for freedom to destroy the roads. They ask merely that they be permitted to put upon them vehicles which will insure the greatest public benefit from their operation, which will not only preserve them as physical assets, but insure the greatest long-run return from the sums expended on them.

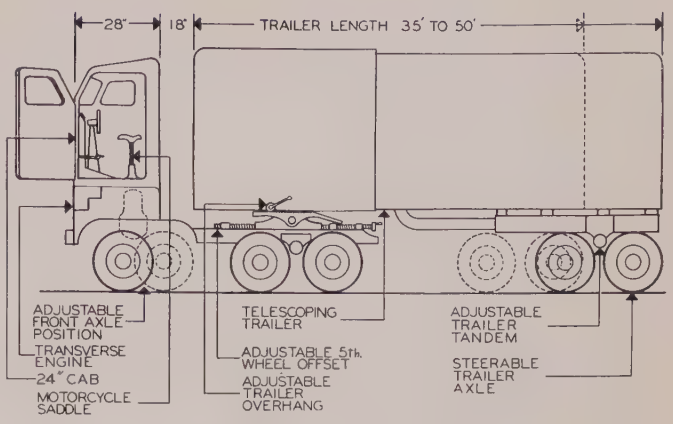


Fig. 11 – Trailer combination designed with adjustable features

Additional References

1. "The Highway User Faces the Law," by A. C. Butler. Presented at SAE National Transportation Meeting, Chicago, Oct. 30, 1951.

2. "Trends in Motor Transport," by V. M. Drew. Presented before SAE Detroit Section, Nov. 1, 1948.

3. "Impact of Size and Weight Restrictions on Motor Transport," by J. B. Hulse. Presented at the SAE National Transportation Meeting, Chicago, Oct. 30, 1951.

4. "Common-Carrier Intercity Vehicle Design Requirements," by W. P. Davis. Presented at the SAE National West Coast Meeting, San Francisco, Aug. 12, 1952.

5. "Let's Build A Truck," by R. C. Norrie. Presented before the SAE Southern California Section, Los Angeles, March 13, 1952.

6. "Transportation Problems in the 11 Western States," by T. V. Rodgers. Presented before the SAE Detroit Section, Feb. 16, 1948.

7. "Problems of the Automotive Engineer," by E. S. Ross. Presented before the Fresno Division of the SAE Northern California Section, Sept. 13, 1948.

8. "Some Engineering Aspects of Future Motor Truck Design," by B. B. Bachman. Presented before the American Association of State Highway Officials, Cincinnati, Nov. 29, 1944.

9. "Why 18,000 lb? Axial Load Effect on Highway Design and Operation," by H. S. Fairbank. SAE *Quarterly* Transactions, Vol. 5, January, 1951, pp. 123-142.

DISCUSSION ON FOLLOWING PAGES

DISCUSSION

Laws for Some Affect Truck Design for All

– Lewis C. Kibbee

American Trucking Associations, Inc.

MY discussion will attempt to bring out some further thinking with regard to the topics the author has already outlined, with a hope that the second look may present a new facet to the problem.

First, let me say that, in general, I agree with the author that our hodge-podge of state laws are, to a large extent, lacking in either scientific or economic fact. This situation is rapidly changing, however, and though truck operators generally deplore the crazy-quilt pattern, we still don't feel that the answer would necessarily be found by having the Federal Government get into the size and weight picture. Nor was the industry pleased with the late but not lamented "load and dimension limitation" section of the SAE Handbook, which gave a bow to the 18,000-lb axle limit and a page of apology for taking this position. If ever you need to use the antithesis of the simile "easy as taking candy away from a baby," it would be, "hard as taking away a 22,400-lb axle from a New England truck operator."

In all fairness to the much maligned legislator, I ask you to put yourself in his position for a moment. He is trying to deal with a complex matter which is only one of hundreds of subjects that come before him. From the welter of ideas proposed, he is asked to come up with a scientific and economic weight law. As a result, the motor carrier industry has been working hard to bring to the various state legislators our desires, based on what factual data we have, and to point out competitive attacks on the industry for what they are. It is a problem which is not going to be solved overnight, but it does appear that regional uniformity may be achievable. Almost every session of every state legislature has some truck bills introduced, and from this constant work comes gradual progress.

In stating his position at the outset, Mr. Horine says ". . . anything which impedes or increases the cost of transportation harms the public." In general I agree. However, I point out that part of the cost of motor transportation is the cost of provision of the highway and that these two factors, highway costs and vehicle costs, must be considered together. At the same time I don't feel that we should approach the problem with a closed mind and set arbitrary load limits forever in an attempt to fix the highway cost, which is only one variable. As has been pointed out, we should provide highways that can carry the largest loads which can be consistently found. This definition of load limits is perhaps difficult to arrive at, but I am certainly unimpressed by highway planners who make "loadometer" studies of traffic in states with an 18,000-lb axle load limit, and then report only 0.5% of the trucks on that road had axle loads over 18,000 lb, and therefore there is no demand for a highway to support more than 18,000-lb axles. It should be evident that in states where say 22,400 lb is the limit, for example, we find commerce generally taking advantage of the highest axle load allowed. By commerce, I don't mean just the line haul for hire carriers, but all types of operators, private, for hire, farmers, and everybody. Likewise, if more states had axle load limits over 18,000 lb, the volume of trucks built to carry these weights would increase. In other words, the demand for heavier axle loading is already here, let there be no doubt about that. Now the question becomes: How much more does it cost to provide the highway facility to carry X – 1000 lb than it does to carry 18,000 lb? Much research is going on at this time in an attempt to find the answer to this question, but as yet we have not arrived at a solution.

Nobody realizes better than do the truck operators that the cost of providing the needed new highways is the present stumbling block in highway development. I am not going to labor this point with which you are all familiar. Let me merely say in passing that the organized trucking industry is on record as accepting its responsibility to pay its fair share when this share has been justly determined.

This brings us to the Maryland road test, and so much has been said about this project that I hesitate to add any more fuel to the fire. However, I agree with the author in his statement that far from proving that 18,000 lb should be a top single-axle limit, and 32,000 lb the tandem-axle limit, it opened the door to a serious investigation as to just how we could have been so wrong all these years. Perhaps as time clears the smoke of battle of this test, we will all concede that it became a turning point in highway design which led us to a true search for facts.

Mr. Horine has given us a very enlightening discussion of the evils of the bridge formula, but one aspect of it he has not touched upon. The American Association of State Highway Officials Code, upon which many state laws are modeled, does not say that the load-carrying allowance of a vehicle combination is based upon the length in feet between the first and last axle of the vehicle or combination, but rather the carrying allowance is based upon the wheel base of any group of axles in the combination. To illustrate, please refer to Fig. 7. Here you will note that, going by axle capacity only, the combination should be allowed 73,000 lb gross. Since it has a wheelbase of 40 ft 8 in., it would be allowed 61,580 lb under the AASHO bridge formula over the first and last axles. However, I gather that it is 31 ft between axles 2 and 5, which under the "any" group interpretation would limit the load on these axles to 53,490 lb, or 26,745 on each tandem. Thus, to get the maximum 61,580 lb allowed under the first and last axle interpretation it is necessary to get over 8000 lb on the front axle when only 26,745 lb is allowed on the drive tandem. With many types of equipment it is difficult to get more than 6500 lb on the front axle with such a light loading of the tandem, and as a result the total gross under the "any" group interpretation would be in the neighborhood of 60,000 lb gross rather than the 61,580 under the first and last axle interpretation or the 73,000 gross if axle loading were the only limit.

Mr. Horine has said that the restriction on the number of vehicles in combination is "of minor importance," and on this point the trucking industry is not in full agreement. One of the combinations with the greatest potential is the tractor, semitrailer, and full trailer; a three-vehicle combination prohibited by the AASHO Code. As this combination can have a 48 ft loading space in 60 ft of overall length, by using two 24 ft semitrailers, it gives the flexibility of semitrailer operation and the loading space of a truck and full trailer. At the same time it does not have the disadvantage of holding a power unit during loading and unloading time, as is the case with the full truck. I believe you will see a lot more of this type of combination as it offers many operational advantages.

Mention has been made of the engineering advances that have been forthcoming to tailor trucks to our weight laws.

For the over-the-road carrier who operates in the upper limits of the weight and length laws most of the time, I believe we have come a long way. However, I sometimes wonder how the other 80% of truck purchasers feel about the distinct trends. Advanced front axles meeting the bridge formula may be a help on a road tractor, but a concrete mixer truck would be better off with the axles set back under the engine to provide a shorter turning radius and provision for longer front springs. The bumper set right up against the radiator may be good when being measured at a state inspection station, but it is useless in the Washington Market section of New York City, when a truck backs into your radiator and disables you at the first whack. In one manufacturer's new models, in the rush to take the space out of the front end, the farmer and the logger and others who operate on and off the highway have been completely forgotten. This manufacturer has put the radiator shutter so close to the radiator core that any collection of leaves, straw, or other trash blown against the radiator will render the shutter inoperative. Fan shrouds have had to be removed, probably at the expense of good cooling; in fact, I doubt if you can remove the fan or fan belts in some models without first removing the radiator core. Thus, state laws which affect a few have changed truck design for all, and inevitably some will have to suffer the consequences. Perhaps someday the farmer will wake up to the fact that he is a big truck buyer and will demand a truck to suit his needs also.

With regard to width, the trucking industry is in full accord with the desire for 102 in. expressed by SAE in 1941. Not that we feel that there is anything holy about tires required for various axle loads, but we would like to have more chain clearance, wider frames, and better tire spacing. Mr. Horine makes me wince as he even suggests springs under the side rails; I operated some dump trucks built like that at one time, and God forbid that anybody ever have to do it again.

The trucking industry most certainly goes along with the desire to protect our highway system. When looking at the taxes an over-the-road trucker pays each year, it is easy to see that in a period of a few years he has more investment in the highway than he has in his rolling stock — why shouldn't he be interested in highways? Ted Rodgers, ATA's first president, put it another way. When he was accused that trucks were tearing up U. S. 66, his quick reply was, "Well, you should see what 66 is doing to our trucks." We realize that there must be size and weight laws, but we ask that they be economic and fair.

The Public Does Benefit, But More Data Are Needed

— Hoy Stevens

Bureau of Public Roads

THERE are as many differences of opinion among highway engineers as there are among automotive engineers. For example, Mr. Horine is known among us SAE'rs as the "heavy-metal man." In his paper he still supports the thesis of the heavy metals, whereas other automotive engineers are greater advocates of the lighter metals, and appear to be having success in their endeavors. However, both Mr. Horine and other automotive engineers are proud of their work and proud of their progress in the development of vehicles during the last 50 years. Similarly, highway engineers are proud of their work and if I must be honest with myself, they are, by and large, quite justified in their self-appreciation. I may point out, as is frequently done, that there are many highways and pavements that have been built and are still in use after 20 years. How many trucks can claim the same?

Enough of this sparring, let's take a better look at the meat of Mr. Horine's paper. As he has done early in his paper, let me be equally frank and say that I think, in the abstract and as a whole, that size and weight limitations have been a benefit to the public. Certain of the size and weight limitations may not accomplish all of the aims that Mr. Horine has outlined, but these limitations are the ones that are in error, not the principle of regulation.

The building of highways is an engineering art, as is the building of trucks. Neither is an exact science, since neither group of engineers has all the facts concerning all the variables that may enter into their respective designs. You automotive engineers are aware of some of the terrific flops that have occurred in truck designs, and that have compelled redesigning. The same thing has occurred with highways, and the highway engineers are aware of this and are searching for new facts.

The road test at Malad, Idaho, being conducted by the Highway Research Board for the Western Association of State Highway Officials, is an example of such basic research work by the highway engineers. The truck and trailer manufacturers and the oil companies are joining in this research by provision of equipment and fuel. The projected Illinois road test of the American Association of State Highway Officials is another example of such basic pavement research. All interests should participate in these research programs in order that they may accept the results as research findings.

The Maryland road test, of which many of you have heard very derogatory remarks, established facts that have seriously disturbed the highway engineers. From this test, the element of frequency of application of heavy loads was definitely shown to be a factor that must be considered in the design and construction of pavements.

In the Maryland road test, a pavement, which had been undamaged under moderate usage by axle loads of the design weight, began to show severe distress when the frequency of applications of design weight axles approached high-density traffic. This is a basic fact from the Maryland road test, and is a fact that is of real concern to the highway engineers. Hence, there is need for the WASHO and the AASHO road tests, which I mentioned earlier, to throw additional light on design criteria.

I think we all can agree that an automotive engineer must have specifications before he can begin to develop a design intelligently, and even then he needs a lot of test work to determine whether his design is adequate to meet the specifications. The size and weight limitations and the bridge formulas, which Mr. Horine has mentioned, have served as such specifications for the highway builders and, in this respect, have been of great benefit to the public.

Mr. Horine makes a case, in glittering generalities, against certain size and weight limitations. He discusses vaguely the reductions in direct transportation costs as the size and weight of vehicles is increased, mentioning also that there may be an upper limit of weight above which there is no further reduction in direct costs. He acknowledges that highways of increased capacity cost more to build, and that the sum of trucking costs and highway costs represents the overall costs of the highway transportation of the goods and commodities that the American people use in their everyday life.

He also appears to advocate a greater contribution to the costs of higher capacity pavements by passenger-car users, on the basis that greater tax contributions to highways will reduce their direct costs of the commodities they buy and use. Possibly some economists may agree that there is validity to this premise. But can it be sold to the general public? In keeping with this thinking, a sales slogan for an increase in the gasoline tax might be, "An X-cents increase in the gasoline tax for passenger cars will decrease the cost of your groceries. Vote for the gasoline tax increase." Will that idea sell?

There is a third factor in regard to size and weight limitations that Mr. Horine has not mentioned. This is the question as to what do the commodities want in the way of size

and weight limits for their transportation over the highways. Stated another way the question is, "What commodities, because of either their very heavy density and/or their minor importance to the economy of these United States are not entitled to visibly full loadings in the cargo bodies of highway freight vehicles?"

Let's take a couple of examples. Frozen orange concentrate weighs 50 lb per cu ft in the can in the carton ready for loading in a refrigerated semitrailer combination. A 35-ft semitrailer has approximately 1800 cu ft of stowage space. The tare weight of a tractor semitrailer combination for this job will weigh about 30,000 lb. The payload will weigh 90,000 lb. Gross load on five axles is 120,000 lb.

You may consider orange concentrate a luxury which should pay a penalty in transportation costs by not being allowed visibly full loading in such cargo bodies. What about wheat? It weighs 42 to 45 lb per cu ft. Other canned goods, wet packed, weigh about the same as orange concentrate. As an extreme, take the case of platinum weighing 800 lb per cu ft. Here, we all can easily agree that visibly full loading in a 35-ft semitrailer is unreasonable. Platinum is a valuable commodity, important to certain parts of our economy, but the total volume is very small, and it can afford to pay a premium in transportation costs by not being permitted to be carried in visibly full body loadings. How many other commodities are of this nature?

We simply do not have the facts about commodity transportation requirements, at the present time. Likewise, we do not have the facts about truck operations costs, as related to the great variety of trucking services, nor do we have the costs of highways to carry a wide range of axle and gross loads, especially in the higher weights, which might be desired by some of the commodities.

The Highway Research Board has a project committee that is working on these three phases to develop factual data that can be considered jointly by all interested parties in order to develop a workable and acceptable compromise for future highway size and weight specifications. The answer will be a compromise because is is unlikely that the three factors will meet in a common point, such as Mr. Horine showed in one of his illustrations. We have no idea as to what such a compromise may be, and will not have until the three groups of data, with their trends, are brought together. I can be rather sure of one thing, however, such a compromise cannot suit everybody.

In the interim, and in connection with this research, we may be visiting some of you to obtain some segments of facts that you may have and that bear on the problem. I trust you will really open up your books to us.

Commercial Vehicles Need Public Acceptance

— Emil P. Gohn
Atlantic Refining Co.

THE following comments might well come under the category of "Preventive Legislation." The growth of automotive highway transportation in this country has been nothing short of phenomenal during the relatively brief period of its existence. Of the 120 million motor vehicles produced since the turn of the century, there are now registered in active use better than 52 million cars, trucks, and buses. This achievement of economic prosperity is one of the highlights of modern civilization, and its rapid development was encouraged to no small degree by the fact that we **had** the best and most extensive highway system in the world.

The history of automotive transportation, up to and including World War II, reveals that there was a concentrated effort devoted to mass production and technical development. During this period, we experienced better and more reliable engines, transmissions, bodies, tires, and chassis. Summed up briefly, industry provided vehicles of progressively higher quality and declining real price, as well as better and cheaper gasoline and tires. This technical phase will continue, encouraged by American ingenuity and mass production know-how.

With the ending of World War II, highway transportation assumed a complex pattern involving many activities associated with its operation. A second phase of highway transportation is now in evidence, and one which, it appears, will exert an equal, if not greater, impact on automotive use in the future. This might be aptly described as the "Regulatory and Taxation" phase. Automotive transportation finds itself in the same position as every business today, namely, operating under continuous and ever increasing legislative controls. This trend was recognized a few years ago when it became apparent that restrictive and other penalizing legislation, from Federal down to local ordinances, was being introduced every year in greater numbers. As a matter of fact, about one out of every four legislative bills introduced in the states affects highway transportation. The accelerated activity is already in evidence in the 44 states now in legislative session, where considerable attention is being devoted to vehicle sizes and weights, together with proposals for increased highway user taxes. The latter includes further increases in fuel tax rates in many states, higher registration fees, and weight distance levies in others.

Maintenance and operation procedures have been aggressively pursued during the past 15 years, with the result that they have been pretty well established to the point of day-by-day operation. It seems that in the overall problems of operating costs, further reductions in transportation costs through refinements in maintenance and operation procedures are becoming increasingly difficult to realize. On the other hand, liberalizing vehicle laws on the statute books can have considerable effect on promoting increased highway use and corresponding reduction in operating costs, while the introduction of restrictive legislation could result in prohibitive operating costs.

We now arrive at a very simple but important deduction, which is if you, as an owner, or automotive fleet operator, or both, do not make it your business to become active in legislative developments, then you must be content to let this business be handled by others. This latter conclusion will lead to results now being experienced by competent business heads where they find themselves in the position of operating under restrictive laws and wondering "What happened?" In an eastern state, there was introduced a bill recently which would have drastically reduced the efficiency of a now popular type of semitrailer unit. Had this bill passed, it is estimated that highway transportation costs in that state would have been increased 30%. Thus, by a single stroke of the pen, the savings resulting from years of maintenance refinements and operating techniques would be wiped out.

The introduction of legislative measures is motivated by a number of reasons, which could be classed broadly under the category of those that are desirable for the general welfare and those that have as their aim the restriction of competition. In respect to the first type, we in the highway transportation field have certain obligations to meet. Failing to meet certain conditions adequately makes us subject to that old slogan, "There ought to be a law....." I might add that the manufacturer as well as the user has a stake in this problem, and it is of little avail to pass a problem lightly around. For example, let us consider some of the problems facing us which, in my opinion, should be attacked vigorously now before legislation is asked to do something about it.

Our commercial vehicles are sharing the highway facilities with Mr. John Q. Public's passenger cars. This places us right into a public relations activity. It can be assumed that the public recognizes the fact that commercial vehicles

cannot compete in performance with passenger cars, and also that commercial vehicles are larger because of the job that they are required to do. However, the public is annoyed by truck noises, particularly those originating at the exhaust. The gradual increase in size and horsepower of commercial vehicles has focused considerable attention on exhaust noise, and it is questionable if the design of mufflers has kept pace with the increased horsepowers. The sound level of commercial vehicles must be reduced. This problem will not be solved if the ball is passed among the vehicle manufacturer, the muffler manufacturer, and the operator. It will require the cooperation of all concerned to offset the passage of restrictive legislation on this subject which might be so worded as to penalize all of us in transportation.

It is quite apparent that this problem is of a magnitude to require cooperative efforts of all concerned. Until such time that answers are forthcoming, it becomes our duty as operators to continue the necessary muffler maintenance and thus keep the exhaust noise level at least equal to that of the vehicle when purchased.

Most states recognize the existence of the problem and have statutory provisions covering engine noises and fumes. The statutory provision reads as follows: "No person shall drive a motor vehicle on a highway unless such motor vehicle is equipped with a muffler in good working order and in constant operation to prevent excessive or unusual noise and annoying smoke."

Variances in the laws of the states are not basically different, and hence are not quoted; however, they all lack any effective means for determining "Is this vehicle too noisy?" The problem, because of the number of variables involved, is not easy; and an effort is now underway to establish, if possible, noise levels that might be considered satisfactory.

Another item that is receiving attention by legislative bodies is the matter of mudflaps. Although the Uniform Vehicle Code does not contain any provisions regulating or requiring the use of mudguard equipment, there are, nevertheless, 12 states requiring mudguard equipment. In addition, a number of states are considering such proposals.

Without attempting to question the advisability of such statutory laws, it is interesting to note that the states involved all agree on the objective desired, namely, to minimize spray of water and other substances from the wheels of the vehicles; however, the outstanding feature of these statutes is the complete absence of uniformity. It is this latter item which concerns interstate vehicle movement, and you might be interested to know that the Transportation and Maintenance Technical Committee of the Society has this subject under consideration, with the aim of developing some factual information on mudflaps.

The performance of commercial vehicles on our highways is of timely importance because (1) it has a bearing on our vehicle operating costs and (2) it has significance in our public relations programs.

Commercial vehicles, because of their high weight-to-power ratio, suffer by comparison to passenger-car performance. Since we are sharing the highway with other forms of vehicles, the disparity in performance between the commercial vehicles and passenger cars is very much in evidence by the general public. It is also quite obvious that if all highways were of the multiple-lane type, this phase of the problem would require little, if any, consideration. However, since we are far from this goal, the performance of heavy commercial vehicles must, therefore, be viewed in the light of present highways. Although we cannot even approach passenger-car acceleration and hill-climbing performance with our heavier vehicles, the availability of more powerful engines for the same displacement will permit, in many cases, an increase in performance. This might take the form of negotiating some hills in the next higher transmission gear or permit faster acceleration. The resultant higher average road speed will contribute to better traffic flow.

The matter of highway safety should be the concern of all of us. The ramifications of this subject are so many and important that a whole session spent on this theme alone could only cover the highlights. I would be amiss, nevertheless, not to touch on this important phase of commercial-vehicle operation. In this respect, it is gratifying to know that those in charge of most of the truck and bus fleets are aware of the impact of maintenance and its relation to safety, and have well coordinated programs under the guidance of qualified personnel to see that they are carried out. I should like to present data released by the Interstate Commerce Commission, Bureau of Motor Carrier Report. Although the statistics cover those regulated carriers reporting to the ICC, thereby constituting only a small percentage of the total vehicles using the highways, nevertheless, the data are revealing. The report says, "The heart of the mechanical-defect accident problem is the brake. Accidents due to defective brakes contributed to over 40% of all mechanical-defect accidents, and they are increasing at a more rapid rate than any other type. Tire failure accidents and accidents due to engine failures are the next most frequent, although each of these types occurs only one-fifth as frequently as brake accidents." Other types of mechanical defects contributing to the overall picture cover such items as coupling devices, steering, wheels and associated parts, lights, springs, fuel lines, and axles, which all added together are responsible for 96% of mechanical-defect accidents.

It is quite obvious that the mechanical items mentioned lend themselves to a maintenance program, any kind of a program, as long as it is thorough and performed at some regular interval.

The above comments on the importance of preventive legislation on highway transportation are submitted to indicate that the future trends on many items are controlled more or less by us in the highway transportation field. This is particularly true on those matters over which we exercise control, such as preventive maintenance, safety, noise, smoke, and selection of equipment. The service that we render for the economical transportation of people and materials brings us very close to the public, and it is the public that requests legislation when we fail to meet many basic requirements. You can rest assured that the greater the degree of public acceptance of our vehicles on the highways, the less the possibility of having restrictive laws passed concerning our business. I believe firmly in self-policing by taking action for the correction of items which appear objectionable or to the best interest of the public we serve. There is no better way to counteract restrictive legislation than by the removal of the causes for such action.

You can also rest assured that the more interest and activity that we devote to this important phase of vehicle operation, the better the chance of assuring ourselves fair and equitable laws. We, as operators, have worked closely with the manufacturer of the vehicle and equipment, resulting in technical advancements to the benefit of all concerned. Equal benefits in the laws, affecting our operation, can be achieved by working with those responsible for the passage and administration of such laws. Remember that your representative is not an expert on most matters proposed by him. He must be guided, for example, on highway transportation matters by reliable information, and because we are intimately connected with such problems, we should be the ones to supply it.

The industry has done an outstanding job in its driver training programs, with the result that the driver of the commercial vehicle is now recognized and accepted by the public as a capable and courteous operator. Our big job now is to have the vehicle accepted on the highway by the public. It is a long range program, and we can help to

reach this goal by diligently attacking some of those items which constitute points of irritation as I have outlined.

Engineers Should Determine Restrictions

– Ernest P. Lamb
Chrysler Corp.

ALL legislation should be predicated upon public benefit. Some of the fundamental matters to consider are protection to life and property, public safety, and the common necessity. Legislators are, then, servants of the public. But so are engineers, and they, by virtue of their special qualifications and experience, are able to judge what regulation is just and necessary to assure maximum highway utilization without undue risk.

Transportation is the keynote of the national well-being, and highways have within the last 25 years become of age. It is a plain fact that failure of this link would be equivalent to complete breakdown. Without trucks, factories could not operate, mines would close, the farmer's produce could not be moved; in short, there would be a collapse of domestic distribution. It is also equally true that legislation which prevents and retards full development of truck transportation is simultaneously throttling the entire economic development.

Discriminatory taxation, for example, designed to relieve the economic pressure on more marginal operations, serves only in increasing the ultimate cost to the entire public. As the scale of an operation increases, the unit cost of the service rendered generally becomes sharply reduced. Trucking is no exception, but unrealistic legislation is preventing the natural economic laws from gaging the optimum size of highway transportation units. The confining barriers are entirely artificial with little regard for the best national interest. Furthermore, present laws admit no greater expansion.

The argument that the highway system cannot bear heavier loads has frequently been challenged; but even if accepted in part, any lack in present adequacy should not be perpetuated. The continued growth and welfare of the country require more and better roads, but development of the road system has certainly lagged acutely behind the demand for more efficient public service.

There is good evidence that roads which will successfully stand up under the destructive effects of the natural elements will also suffer little added deterioration from the effect of wheel loads of at least 9000 lb and perhaps more. In some localities, axle loads over 22,000 lb have been in successful use for many years.

Beliefs concerning prohibitive cost of roads designed for more than 18,000-lb axle loads are not confirmed by the experience of 12 eastern states where 20,000-lb or over axle loads have been permitted for years. In studying road costs, it becomes evident that many other matters beyond the paving expense are vital.

A very large fraction of cost is reflected in acquiring the right-of-way, grading, draining, guard rails, road signs, and the like. In many cases, no added expense is required to withstand heavier loads, and more important, higher outlay can easily be justified in more economical transportation for all people. Truck operators furnish an indispensable service for which all consumers of goods and services must pay. The better the system, the lower in the long run will be the proportionate cost to the entire American public.

Axle-load limits are only one of several hampering restrictions. In most states the load on a pair of axles is less than allowed on each axle in the straight truck. Today no sound reason exists for limiting bogie loads by other than the permissible individual axle weight.

Many lawmakers have given special attention to the protection of highway bridges. Bridges do differ from roads in that the pavement is not carried on the subsoil, but on manmade beams and girders. Bridge engineers have set up formulas for capacity. Just as there are many differences in bridges, there are also many variations of the formula. Unfortunately there are many states where a formula, or a table derived therefrom, is a part of the law. The law applies to all vehicles found within the state and as far as the operator is concerned is restrictive even if he never crosses a bridge.

Legislators control not only the weight but also the physical dimensions and arrangement of the vehicle combination. Overall vehicle height is perhaps the one dimension about which there is the greatest satisfaction, the least cause for criticism, and the best justification. Clearance at practically all important underpass locations is not less than the maximum height limit, and it is unlikely that any new construction will anywhere be less. Eventually, as conditions of traffic warrant or reconstruction becomes necessary, all existing low-clearance conditions will be corrected.

When it comes to vehicle length, what combinations may be operated, and the permissible length of combination, there are numerous important differences in the various states. As it stands today, there are often such conflicting requirements between neighboring states that interstate operations are very closely curtailed. As the scope of an operation increases, embracing more states, so does the difficulty.

The operating difficulties are about analogous as they would be for the railroads if each state had its own ideas as to the size of locomotive which would be proper and the number of cars which would be permitted in a train.

The trucking industry has been forced to develop within a web of growing restrictions. Designers have sought to give their vehicles the widest possible adaptability. Some special types have been developed to meet particular length restrictions. These efforts have been directed towards getting maximum payload within the permitted space and weight limits.

The forces tending towards the COE design have been incapable of being reduced to a common denominator. For most applications the better accessibility, lower cab height, lighter weight, lower cost, greater comfort, and superior ease of entrance have strongly favored the conventional type. Only where the ultimate in payload space for overall length is the deciding element does the COE win out. In the final analysis, there are but few applications where the COE design would be the logical selection if it were not for the artificial restrictions which do not permit the operation of conventional vehicle combinations of adequate length to fill the economic demand.

Restrictions in weight, axle location, length, and height are not the whole story. All but three states say that the overall width must not be over 96 in. This is practically the same as saying that axle load must stay at 18,000 lb. Greater axle loads are necessary for further development of truck transportation and a 102 in. width is needed to permit adequate tires for these increased loads.

The discussion has summarized the conclusion that present size and weight restrictions are limiting the development of highway transportation. Further, the dependence of the entire nation on trucks has reached the stage where any retarding effect on truck transportation has similar effects on the entire national economy.

Some highway studies have been completed and others are under way to obtain data for a more realistic approach to these problems. For the benefit of the public there is a need for:

1. Uniformity – The same vehicle or combination with identical loads on all through highways of the country.
2. Length – A minimum combination length which will permit the use of conventional equipment.

3. Width – Overall body width of 96 in. with 102 in. over the rear tires.

4. Height – A uniform overall height limit of 13 ft.

5. Bridge formula – Modification or removal of these restrictions.

6. Axle spacing – Removal of these restrictions.

7. Axle capacity – A uniform permissible load of approximately 22,000 lb per axle.

8. Gross vehicle or combination weight – Limitation by number and capacity of axles.

Uniform Limitations Are Highly Desirable

– Fred B. Lautzenhiser

International Harvester Co.

UNIFORMITY across the country among the various states is highly desirable. Truck operators on the West Coast are restricted to 18,000-lb axle loads, yet with the overall length restrictions they can attain approximately 77,000-lb gross combination weight. They are quite happy with it. I believe that the whole country would gain if they would work harder for uniformity or for a minimum-maximum as set up in the AASHO Size and Weight Code.

It is not likely that we will know the answer to the size and weight problem until after the completion of Highway Research Board Project No. 5, "The Economics of Motor Vehicle Size and Weight," and the phase of it that will be constituted by the AASHO contemplated road test.

Use Trucks for Short Hauls, Railroads for Long Hauls

– F. C. Kreitler

General Electric Co.

IN his paper Mr. Horine offers a number of constructive ideas—for example: the standardization of size and weight requirements between the various states, the improvement of truck design as related to government restrictions, and the need for roads adequate to carry present-day traffic. But, while he suggests that, "it is therefore most appropriate that we as engineers approach a consideration of the effect of vehicle size and weight regulations from the standpoint of the people at large rather than from our own narrow viewpoint," the author seems to take a narrow look at the broad transportation problem. Therefore, certain parts of the paper may prove confusing to those not closely connected with the transportation industry.

Mr. Horine states without qualification that, "trucks haul more tonnage than all other forms of transportation combined." When one includes all farm vehicles and delivery trucks handling goods locally, this statement is correct. It is in this smaller short-haul service that the truck in total performs its primary and biggest function. In his paper, however, the author confines himself to a discussion of the intercity service where size and weight restrictions are encountered. In this sphere the truck performs a secondary and less important function for the general public. Statistics published by the Interstate Commerce Commission and the Public Roads Administration shows that, on a ton-mile basis in 1951, trucks – private as well as contract and common carrier – hauled only 12.4% of the total ton-miles. In 1950 the figure was 12.5%, and in 1940, 8.4%. Moreover, only a fraction of this traffic is moved in large trucks impeded by government restrictions.

The difficulties of big over-the-road trucks seems to be economic rather than legislative. The general public, through size and weight limitations, is asking only that the relatively few big trucks operate within the framework of roadway provided for the vast number of private autos and small trucks.

From the general public viewpoint bigger and heavier trucks seem to have questionable long-range economic advantage for day-to-day transportation of goods between large cities. Between big cities where the majority of big truck tonnage moves, the railroads already provide high-capacity, heavy-load arteries on their private rights of way. If government restrictions were the only problem – and if big intercity trucks are as vital to our economy as Mr. Horine implies – then it would seem logical for truckers to build private roads between major points and thus be able to determine their own restrictions. An alternative that appears to be growing increasingly attractive is to combine the superiority of the truck for short-haul, pick-up, and delivery service with the ability of the railroad to provide economical long-haul transportation. From the standpoint of the people at large such cooperation makes much better economic sense than bitter competition that may result in the duplication of tremendously expensive long-haul facilities.

Author's Closure To Discussion

I am indebted to Mr. Kibbe for his correction of my conception of the bridge formula. It is even worse than I thought. No wonder most of the states using it resort to a tabulation! His remarks undoubtedly reflect the attitude of over-the-road operators in general and as such are most appropriate to this discussion.

As usual, Mr. Stevens has contributed constructively and informingly to the subject under consideration. His outline of research projects now planned and in process, and particularly the objectives in view, is most encouraging. That's all we want – the facts. We would have no objection whatever to size and weight limitations if they were based on facts, current facts, and if sufficient flexibility were retained to enable them to keep pace with future developments.

I must agree when he says that size and weight limitations **in the abstract** might be of benefit to the public; that the principle is sound, even though the specific limits are in error. Unfortunately, the public is being forced to suffer from the effects of the restrictions as they are and therefore cannot enjoy the benefits that might accrue were they more realistic and equitable.

I like Mr. Gohn's statement: "There is no better way to counteract restrictive legislation than by removal of the causes of such action." In the spirit of this thought, it might be profitable to determine what these causes are. Are they:

Improper design of vehicles?
Heedless operating practices? or
Inadequacy of roads and bridges?

Perhaps all three have combined to bring about the enactment of present limitations.

Until the facts that we so sorely need are determined, I think, in general, Mr. Lamb's prescription would be a safe guide for new and amended size and weight restrictions. However, I still contend that, so long as there are such vast sectional differences in terrestrial formation, climate, and density of population as now exist in the United States, regional, rather than country-wide uniformity is desirable.

Mr. Lautzenhiser has put his finger on what might prove the most significant research project bearing on this problem, now in process – the economic study. After all, what we have been discussing here is basically an economic question. So, in closing, I should like to repeat: Equitable size and weight legislation is that which fosters the most efficient use of the combination of the vehicle and the highway, to the end that the public will receive the maximum return on its investment in highways and bridges through convenience, celerity, and economy of transportation.

DESIGN

EXPERIENCE with fighters and transports during the past two years has emphasized the importance of designing for fatigue in future aircraft.

Fifteen years ago, airframe fatigue problems were limited to detail design considerations in which the designer simply avoided unusual stress raisers by providing generous radii and smooth continuity in the structure. By following these rules, primary airframe fatigue problems were minimized, except in the few cases where good design practices were violated. Since then, a gradual change has taken place. Aircraft are expected to last longer, and today some transports such as the DC-3 have operated 50,000 hr or more. Fighters and trainers have experienced service of over 3000 hr of the most severe abuse pilot training can give.

New materials such as 75S and 78S have been developed that invite static design tensile stresses far above the 24S-T value used in these earlier models. Refined stress analysis methods and accurate in-flight load measurements have eliminated many conservatisms inherent in previous designs. These factors add up to the conflicting requirements of longer life from a lighter and less conservative structure operating at higher stresses.

The tensile strength of available aluminum alloys today has increased about 50% over the 24S-T allowables of 1938. At the same time, however, the fatigue S-N curves for these materials has not improved, and if the higher strength of new materials

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STEPS in fatigue design of aircraft lower wing surfaces, as presented here, are:

1. Selecting a structure that permits easy fatigue-crack detection. X-ray inspection of blind areas is suggested.

2. Setting a design quality standard. This is the important stress-concentration factor K_t, which is shown to have a lower practical limit.

3. Choosing a design gross area stress rather than net section stress to give the desired life. The gross area stress is also a better basis than the 1.0g operating stress level for comparing fatigue life in similar aircraft.

4. Step testing to check design quality. This preferred method of fatigue testing is discussed here.

The Author

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FOR FATIGUE

E. H. Spaulding, Lockheed Aircraft Corp.

This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 1, 1953.

is utilized and design details remain the same, there is a very large reduction in fatigue life.

Most gusts and maneuvers will cause stress changes above the so-called endurance limit of aluminum alloys. For this reason, fatigue cracking will occur sooner or later in any well designed airframe if operated for sufficient time. The engineer must build into the structure enough life to take care of normal expected needs, but operators can expect fatigue difficulties if their aircraft are subjected to frequent abuse beyond normal expected conditions.

In the interest of limiting the length of this paper, the discussion will be confined to wing lower tension surfaces.

Important Concepts

Fatigue Properties of Aluminum Alloys – Two important facts concerning basic material fatigue properties for aluminum alloys are presented by Templin, Howell, and Hartman.[1] These are:

1. "The fatigue strengths determined in the transverse directions are not significantly different from those for the longitudinal direction." (The authors based this statement on extensive tests conducted in the three directions: with grain, long transverse, and short transverse.)

2. "When comparisons of notch fatigue properties are made on the basis of scatter bands rather than on the basis of the data on individual lots of material, there seems to be no clear-cut, consistent advantage for any one of the three alloys." (The three alloys in this case were 14S, 24S, and 75S plate, extrusions, and forgings.)

What these investigators have found to be true for 14S, 24S, and 75S plate, extrusions, and forgings appears also to hold for all common structural aluminum alloy products. For practical purposes we can use one *S-N* curve to represent all structural aluminum alloys in any form and in any grain direction when subjected to a given type of loading and ordinary stress concentration even though their ultimate strengths may vary considerably.

Operating Stress Level – The 1.0*g* operating stress level is not a good basis for measuring the relative fatigue life of two similar aircraft types except when they are designed to the same gust-load factors.

When discussing fatigue life of a particular airplane, many have asked "What is the 1.0*g* stress level?" The inference is that a high 1.0*g* stress level results in a shorter fatigue life. Fig. 1 is a plot of varying stress versus mean stress at a stress-concentration factor of 4.5 for two values of constant life, 50,000 cycles and 2500 cycles. Assuming two airplanes to be designed for a limit gross area tension stress of 40,000 psi, lines have been drawn representing the stress change due to 5- and 15-fps gusts. It is seen that the 4.0*g* and 2.5*g* airplanes can both withstand about 50,000 completely reversed cycles of load corresponding to gusts of ± 5 fps. Also, it is seen that the 2.5*g* airplane can take more cycles of a 15-fps gust than can the 4.0*g* airplane.

Longer gust fatigue life is possible in the low-load-factor airplane with high wing loading and a higher 1.0*g* stress level if the design gross area stress is the same for the two being compared. A sounder basis for evaluating the relative fatigue

[1] See "Variations in Fatigue Properties of 14S, 24S, and 75S Plate, Extrusions, and Forgings, with Specimen Orientation," by R. L. Templin, F. M. Howell, and E. C. Hartman, Aluminum Co. of America, Feb. 7, 1950.

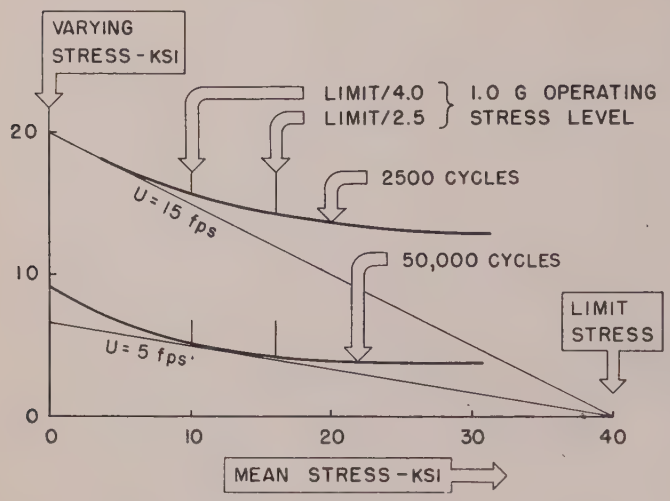

Fig. 1 – Effect of 1.0_g stress on service life of two aircraft designs to same limit stresses

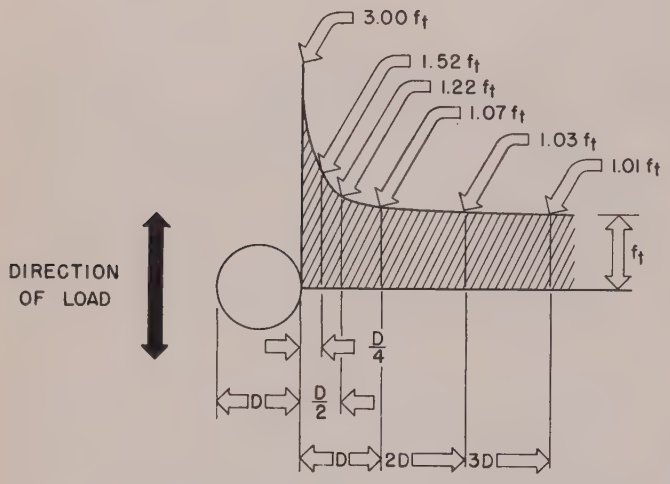

Fig. 2 – Stress distribution in an infinite sheet pierced by a single hole (from Timoshenko[2])

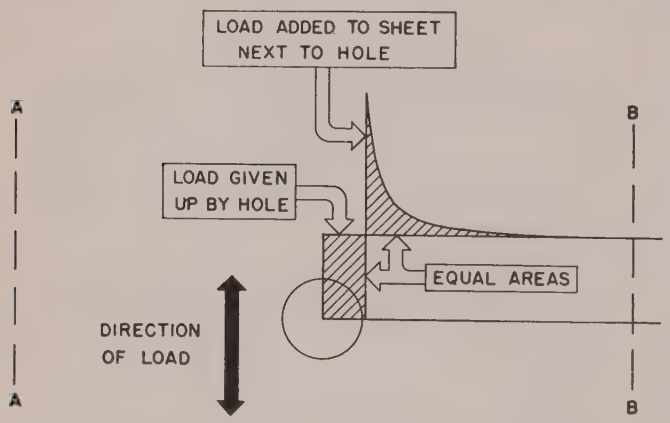

Fig. 3 – Representation of load exchange when hole is placed in an infinite sheet

life of two airplanes is to compare the limit or ultimate gross area stress levels because the most damaging gusts are in the order of 5 fps, and it is necessary to reduce the limit stress to permit more of such loading cycles.

Gross Area Stress as a Fatigue Life Index – The elastic stress distribution at a single hole in an infinite sheet loaded in tension has been determined[2] to be as shown in Fig. 2. If this sheet were trimmed along lines *AA* and *BB* of Fig. 3, there would be a negligible effect on the peak stress because the part being trimmed away has been relatively unaffected by the hole in the remaining strip. This occurs because the load that can't go through the hole takes a short detour and only affects the material in a highly localized area. The diagonal shading of Fig. 3 illustrates the load given up by the hole and redistribution of that excess load in the material adjacent to the hole.

Rivets are seldom spaced closer than three diameters apart because the net section efficiency at this spacing is only 67%. When a series of holes are placed in the sheet at 3*D* spacing as in Fig. 4, the resultant stress distribution between holes can be determined by superposition. For example, the peak stress relative to the gross area stress can be approximated by the product of:

The peak stress for a single hole = $3.00\ f_{t_{gross}}$

The effect of the new hole to the right = $1.02\ f_{t_{gross}}$

The effect of the new hole to the left = $1.02\ f_{t_{gross}}$

And the final peak stress = $3.00 \times 1.02 \times 1.02 = 3.12\ f_{t_{gross}}$ which is only 4% greater than for a single hole in spite of the fact that one-third of the material has been removed.

If the net section had been used as a basis for stress calculation, the corresponding value of $K_{t_{net}}$ would be 2.08 which is a 27% change from the initial value for a single hole.

If the holes are closer than 3*D* apart, the designer should be concerned more with static strength rather than fatigue. The low gross area stress dictated by static strength for rivet spacings less than 3*D* will reduce peak stresses to the point where fatigue is less critical.

The above reasoning has shown that elastic-strength-concentration effects are highly localized around a hole and that for reasonable spacings, one hole has little effect on the next. The same reasoning can be applied to holes filled with rivets or screws which load the sheet in bearing. Bearing stresses cause additional stress concentrations to be superimposed on existing peak stresses at the

[2] See Part II of "Strength of Materials," by S. Timoshenko. Pub. by Van Nostrand, New York, 1930.

hole, but this too is highly localized, and similarly, the effect is not felt to any practical extent by the next hole.

This discussion was intended to show that for practical hole spacings the peak stress is primarily a function of the gross area stress, the type of hole, and the bearing stress. Hole spacing has little effect on peak stress within practical limits; therefore, if two similarly loaded structures of high-quality detail design are being compared, the relative fatigue life can best be estimated by comparing the gross area stresses, and differences in net section between the two can be ignored as a first approximation.

Stress Concentrations – Elastic stress concentrations (K_t) have been chosen for use throughout the paper. These values are based on the geometrical configuration and type of loading and are usually evaluated by photoelastic analysis. Values of K_t for various conditions are contained in many standard handbooks and other sources.[3] When it is said that a joint has a certain value of K_t, it is meant that its behavior based on the gross area stress of the basic structure near the joint is the same as a standard fatigue test coupon which has the same value of K_t. Papers by Grover, Bishop, and Jackson,[4, 5] and by Grover, Hyler, and Jackson,[6] are the source of such coupon fatigue data; and for the purposes of this paper, the lower side of the scatter bands are used.

Fatigue tests conducted on a C-46 "Commando" are reported by McGuigan.[7] The 34 failures reported were reanalyzed to determine values of K_t which differ from K_1 and K_2 given in that report. It was found that K_t varied from 2.7 to 7.0 and that only 12 failures had values less than 4.5.

Fatigue tests of many structural joints in current and anticipated use have been conducted at Lockheed. These tests have led to the conclusion that it is difficult to design a joint having a K_t less than 4.5 but that this can be done if exceptional care is exercised.

As a general rule, the designer should not depend on too low a value of K_t in conventional structures as he may find it impractical to obtain.

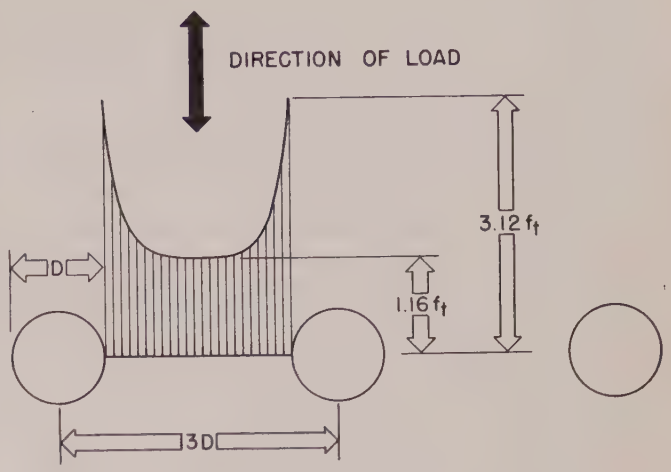

Fig. 4 – Stress distribution between two holes at 3*D* spacing

There is a possibility of depending on a lower value, say $K_t = 4.0$, if adhesives are used or if integrally stiffened machined structure is employed. One must remember that a plain hole gives $K_t = 3$ and that if a rivet or screw loads the hole, K_t will always be greater. Fig. 5a illustrates the variation of K_t with bearing stress in a loaded plain hole of the spanwise splice shown in Fig. 5b.

The values of K_t from Fig. 5a are increased further by nonuniform bearing of single-shear joints and by localized bearing stresses at the bottom of countersunk holes. Raised machined pads which

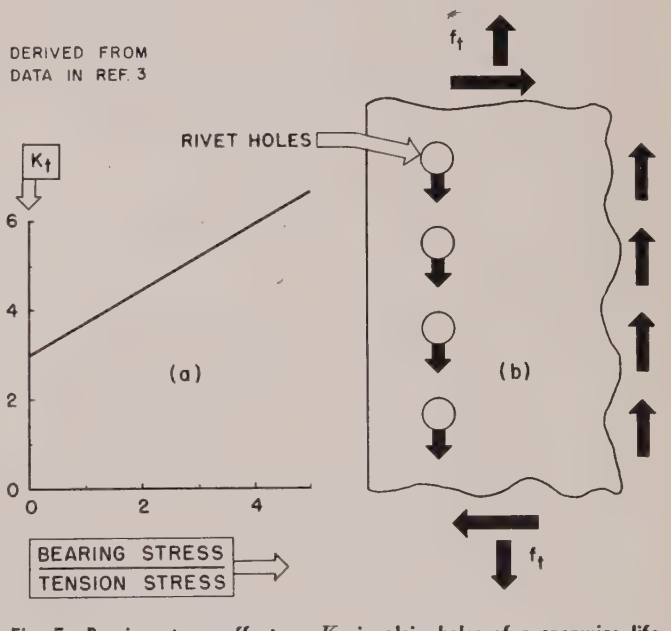

Fig. 5 – Bearing stress effect on K_t in plain holes of a spanwise life in tension material

[3] See Cooper Union Bulletin, March, 1943, "Effect of Material, Loading, and Stress-Concentration Factors on Design," by George H. Neugebauer.

[4] See NACA TN 2389, June, 1951, "Fatigue Strength of Aircraft Materials – Axial-Load Fatigue Tests on Notched Sheet Specimens of 24S-T3 and 75S-T6 Aluminum Alloys and of SAE 4130 Steel with Stress-Concentration Factors of 2.0 and 4.0," by H. J. Grover, S. M. Bishop, and L. R. Jackson.

[5] See NACA TN 2390, June, 1951, "Fatigue Strength of Aircraft Materials – Axial-Load Fatigue Tests on Notched Sheet Specimens of 24S-T3 and 75S-T6 Aluminum Alloys and of SAE 4130 Steel with Stress-Concentration Factor of 5.0," by H. J. Grover, S. M. Bishop, and L. R. Jackson.

[6] See NACA TN 2639, February, 1952, "Fatigue Strength of Aircraft Materials – Axial-Load Fatigue Tests on Notched Sheet Specimens of 24S-T3 and 75S-T6 Aluminum Alloys and of SAE 4130 Steel with Stress-Concentration Factor of 1.5," by H. J. Grover, W. S. Hyler, and L. R. Jackson.

[7] See NACA TN 2920, April, 1953, "Interim Report of Fatigue Investigation of Full-Scale Transport Aircraft Wing Structure," by M. J. McGuigan, Jr.

reduce the stress locally at chordwise joints aid in lowering the effective K_t based on the gross area stress near the joint. A raised pad for a spanwise splice, however, does not lower the tension stress because the added material simply operates at the same strain as the adjacent structure. A spanwise pad, however, does some good because it reduces the bearing stress and minimizes countersink effects. Friction, due to the clamping of rivets and bolts, is helpful as it prevents the smaller fluctuating loads from being felt by rivet or bolt attachments. Friction, however, is hardly to be relied upon in the usual oil-soaked structures of today's aircraft.

Secondary stress concentrations such as surface corrosion pits, die drag in extrusions, scratches, and other manufacturing blemishes are seldom present in the laboratory fatigue-test specimen. These blemishes, however, are frequently present in actual structures and have caused the customers considerable concern. Drake[8] discusses this situation and points out that the life of a structure is controlled by the intensity of the maximum stress raiser and not by the number of them. Blemishes may usually be ignored in common aircraft construction if they are of less severity than the stress raisers designed into the structure. The only danger might be where a blemish exists at the edge of a hole or radii, thus compounding the two stress concentrations. Fatigue tests have been run at Lockheed on specimens with various nominal surface blemishes and having a central open drilled hole. In all cases, failure started at the hole and no difference in life was detected that could be attributed to the surface condition. It is probable that the roughness of the drilled hole was the governing factor and that surface roughness didn't make it any worse.

After considering all the effects tending to increase K_t, the designer would probably be unduly optimistic if he were to rely upon values of K_t below 4.5 even though it may be possible to achieve a lower value in the laboratory.

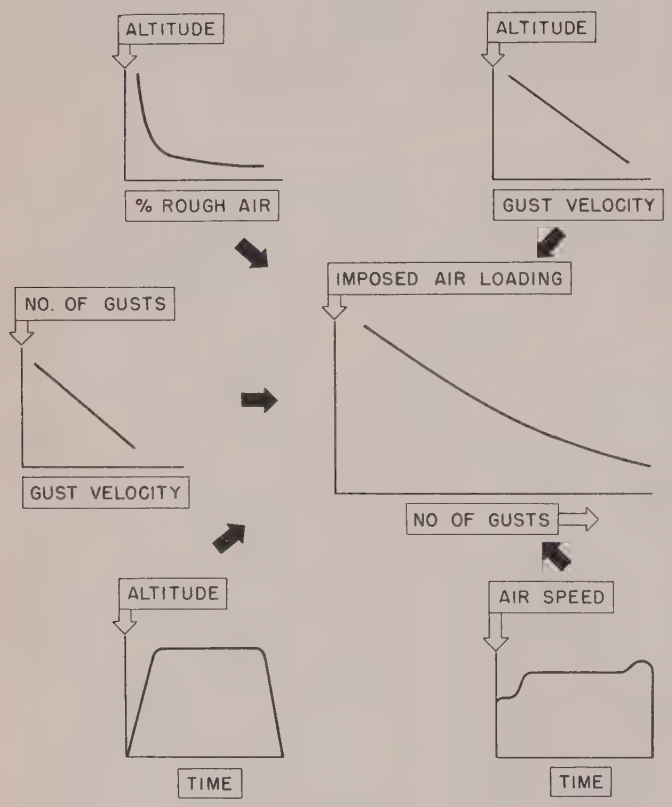

Fig. 6 – Gust spectrum determination

Design for Fatigue Damage Detection

A wing design in which all the bending is taken by a single spar presents the greatest risk or requires the greatest conservatism in design to insure safety equal to a more favorable type of structure.

If a small crack starts in a heavy single spar cap, its rate of progression is rapid, and it can be quite deep without appreciable surface opening and is therefore difficult to detect by visual inspection. X-ray inspection of heavy caps is not satisfactory, and if the cap is hidden by a fitting or doubler, it may crack completely through before detection. It has been demonstrated that a crack through only 10% of the cross-sectional area of a heavy member can reduce its static tensile strength to one-third the original value.

A conventional box structure having many stringers and small spar caps offers the best opportunity for fatigue-crack detection before the crack can develop to a dangerous degree. This is especially true if the wing box is an integral tank, as fuel leaks give early warning of cracking. In structures such as this, even though a crack may start in a hidden area, it has the opportunity to progress into the open and be detected visually before causing serious strength reduction. The thinner sections of this type construction usually permit reliable X-ray inspection of hidden areas. The rate of crack progression across many stringers is much slower than in single spar caps, and there is evidence to indicate that small cracks have flown many thousands of hours before growing to serious proportions.

Sources of Varying Loads

In order to determine the nature of repeated loads to which aircraft are subjected, the NACA,

[8] See *Journal of Aeronautical Sciences*, Vol. 13, May, 1946, pp. 259-269: "Effect of Notches on Static and Fatigue Strength," by D. W. Drake.

Air Force, Navy, CAA, various aircraft companies, and several foreign nations have been gathering statistics on gusts and maneuvering loads for the past 10 years or more. Recent improvements in techniques for obtaining and analyzing such statistical data have resulted in fairly reliable information on gusts and maneuvers. However, even these present techniques require refinement, and there is yet much to be learned before such data can be used with complete confidence. Knowledge of landing and ground-maneuvering loads is scant, and the designer must resort to his good judgment rather than factual data in this field.

In spite of these difficulties, the designer must decide on a loading spectrum and a desired life for the structure he is to design. Some of the things to be considered when choosing a loading spectrum are:

1. The basic low-altitude gust spectrum for rough air.
2. Variation of rough air with altitude.
3. Gust intensity versus altitude.
4. Time versus altitude for a typical flight.
5. Time versus air speed for a typical flight.
6. Maneuvering spectrum.
7. Ground loads:
   a) Landing impact.
   b) Braking.
   c) Turning.
   d) Engine run-up.
   e) Taxi over rough runway.
8. Validity of gust and maneuvering data.

Fig. 6 illustrates the type of information one must work from in order to construct a flight air-loading spectrum. Numerical values for such data are available in NACA literature. Airplane structural details and dynamic response characteristics must be considered when converting air loading into internal stresses.

Fig. 7 shows three sets of gusts which cause progressively more fatigue damage. Fig. 7b is more damaging than 7a because the stress changes occur at a higher mean stress level. Fig. 7c is more damaging than Fig. 7b because the stress change is greater for each bump. These are shown to illustrate the difficulties encountered when interpreting statistical gust data. If Fig. 7b were counted as three gusts of $\Delta n = 2.0g$ as in Fig. 7c, that would be wrong. Also it would be wrong simply to count Fig. 7b as one gust of $\Delta n = 1.5g$ plus three gusts of $\Delta n = \pm 0.5g$ because two of the gusts would then be counted the same as Fig. 7a.

At best, the gust- and maneuvering-load spectrum determination is a complex problem. The necessary simplifications reduce accuracy to the point where one may lose confidence in this approach. The trends, however, are well established; and these combined with actual experience may be the key to future handling of this situation.

Analysis of fighters has revealed that maneuvering loads may produce 95% of the damage and that gust loads can be safely neglected. The opposite is true in the case of transports where gusts produce the major damage and maneuvers produce relatively little.

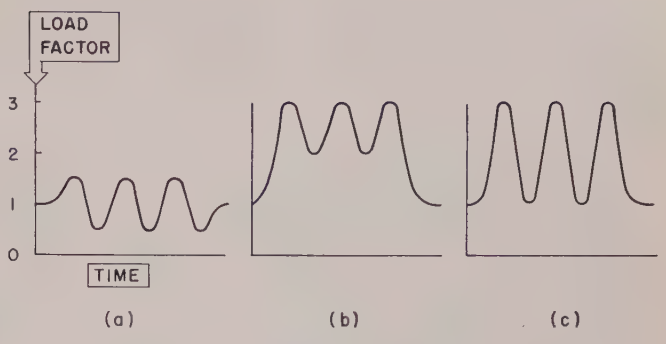

Fig. 7 – Types of load factor versus time curve

Fatigue Testing

It is rather common experience that the state of the art of fatigue-life prediction of structural joints and elements by analytical means is not sufficiently advanced to produce answers with high degree of accuracy. With fatigue becoming a problem of such increased importance, the Lockheed Company has adopted a policy of fatigue testing primary wing tension joints and important discontinuities in the wing lower surfaces. Fig. 8 shows such a joint under test in the Lockheed 500,000-lb fatigue-testing machine. This has been done to insure that the quality of detail design meets certain preestablished minimum standards. As a result of this policy, many joints have been redesigned during the past year subsequent to initial tests. In one case, such redesigns improved the estimated service life of the joint by a factor of 200. These serve to illustrate the fact that in spite of extreme care in the detail designs of a joint, it is not always possible to obtain optimum configurations from a fatigue standpoint without the aid of tests.

S-N curves were obtained for various joints during Lockheed's early experience with fatigue testing. This method of testing was abandoned immediately because of the following difficulties:

1. A large number of specimens were required to obtain any sort of a curve. This was expensive and time consuming.

2. One joint tested gave three different types of failure at widely different locations depending on the load level being used. This suggested that three different stress-concentration factors were present and acting at three different stress levels. The curve that was obtained must have been the lower portion of three *S-N* curves.

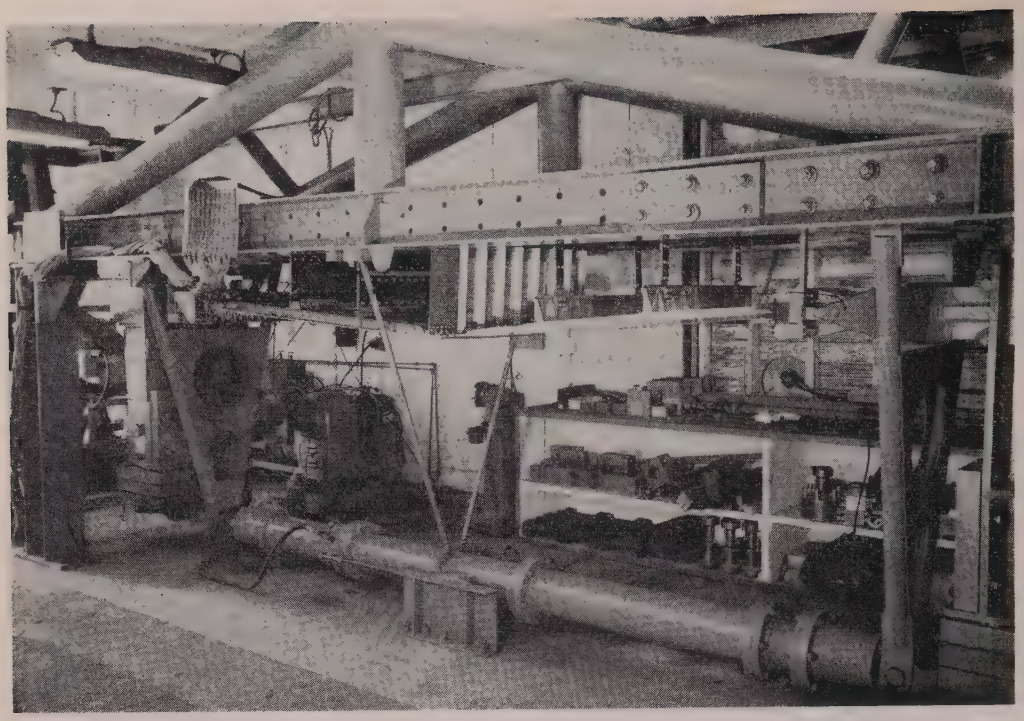

Fig. 8 – Super Constellation wing station 80 joint in the Lockheed 500,000-lb fatigue-testing machine

3. If the Miner theory of cumulative damage[9] were applied to such *S-N* curves, it led to overly conservative answers which were not acceptable.

The next testing method considered was to pick a constant load level and compare structures on this basis. This was abandoned because it was known from past experience that the most damaging load level was not the same for different specimens. Also, this type of testing would always leave a question regarding the effects of higher and lower loadings and the validity of Miner's theory.

"Step" testing was the method finally selected for fatigue testing. Two standard loading spectrums are used: one for transports and one for fighters. The loading-spectrum slope has been chosen to reproduce as nearly as possible the typical slope for a transport or fighter as the case may be. This same spectrum is used for any location in all airplanes of a given class (transports or fighters) with no regard to the actual design stresses. The purpose for such standardization is to allow direct comparisons of design quality for the various joints tested. The actual loading-spectrum slope for similar aircraft types is about the same for practical purposes and small variations in the 1.0*g* stress level have little effect on the number of cycles to failure; therefore, such standardization is possible and involves little error. In this method, the standard loading spectrum is broken down into about 20 steps. The spectrum is made small enough so that it must be repeated about 40 times before failure of a good design. Random loading is thus approximated, and many assumptions otherwise necessary are eliminated.

Fig. 9a represents what is actually done when step testing. Fig. 9b is the normal method of plotting such data so that a smooth curve can be drawn. The actual testing condition is represented by the stepped dashed line in Fig. 9b, but there is little error involved by drawing a smooth curve as there are about 20 steps in a complete test program.

Fig. 10 is a plot of *S-N* curves for aluminum alloys at a mean stress of 14,000 psi and is based on the low side of the scatter band of data from references in footnotes 4, 5, and 6. On the same chart is an example of a transport step test failure curve shown as a straight diagonal line tangent to an *S-N* curve for $K_t = 4.5$. A joint having $K_t = 4.5$ subjected to step testing should fail before the loading-spectrum line becomes tangent to the *S-N* curve for $K_t = 4.5$ shown in Fig. 10. For the case shown tangent to $K_t = 4.5$, the actual value of K_t for the joint must be somewhat less than 4.5. For a transport loading spectrum, the difference between the actual and tangency values of K_t is smaller than for the case of a fighter loading spectrum. A fighter loading spectrum is so nearly parallel to the upper portion of the *S-N* curves that considerable damage is done over a wide range of loading levels. For this reason it is expected that identical joints subjected to fighter and transport step-test loading spectrums will fail

[9] See *Journal of Applied Mechanics*, Vol. 12, September, 1945, pp. A-159 – A-164: "Cumulative Damage in Fatigue," by M. A. Miner.

at different tangency values of K_t even though their absolute values of K_t are the same. The fighter loading spectrum should cause the earliest failure.

After step testing a considerable number of joints and plotting their failure lines on a chart such as Fig. 10, a pattern will be set establishing a standard of quality that can be expected and a goal to achieve in future designs. At Lockheed the goal for transports is set as a tangency value of $K_t = 4.0$ but using 4.5 in life calculations to allow for scatter. At the present time there is insufficient experience with fighter testing to establish a tangency point K_t goal for fighter aircraft.

Fatigue Analysis of Fighters

Maneuvers produce the major fatigue damage in fighters, and statistical data are available from the Air Force and Navy on maneuvering loads measured in training and combat.

A stress-concentration factor should be chosen for the design at the very start and should be the lowest practical value based on the designer's experience. By use of Miner's theory,[9] *S-N* curves for $K_t = 4.5$, and a loading spectrum, the life of a given fighter structure was computed for different values of design ultimate gross area stress. The lower side of the *S-N* curve scatter band was used for this life prediction because, in a well-designed wing structure, there are many possible locations for fatigue failure, and one location will probably have a life on the low side of this scatter band. Data employed in the analysis are presented in Figs. 11 and 12, and the results are shown in Fig. 13. For reference, two successful service aircraft have been plotted to illustrate where they fit on the curve.

Of interest is the lower scale of Fig. 13 which predicts the number of cycles of from 1.0*g* to 7.33*g* the wing can withstand. A structure good for 1000 cycles will have a life of about 1000 hr while one good for 4000 cycles will have a life of about 6800 hr. This illustrates that constant repeated loading is not directly proportional to fatigue life and is further reason for conducting step tests instead. Another point of interest is that a structure expected to withstand 10,000 cycles of repeated load from 1.0*g* to 7.33*g* must have its ultimate gross area stress limited to about 31,600 psi. This would present a serious weight penalty in any design and provide life far in excess of actual needs for ordinary service. The customer should recognize the penalties arbitrary requirements may impose and avoid hasty decisions when establishing test requirements. One must realize there will be a large scatter in actual loading spectrums for two identical aircraft in different service operations. This scatter may be as high as 100 to 1. If one were to design a structure so that no airplane could possibly fail from fatigue, that structure would be heavy and would penalize the performance of all sister aircraft. This unfortunate fact requires the engineer to decide on the degree of conservatism to be used as a design policy.

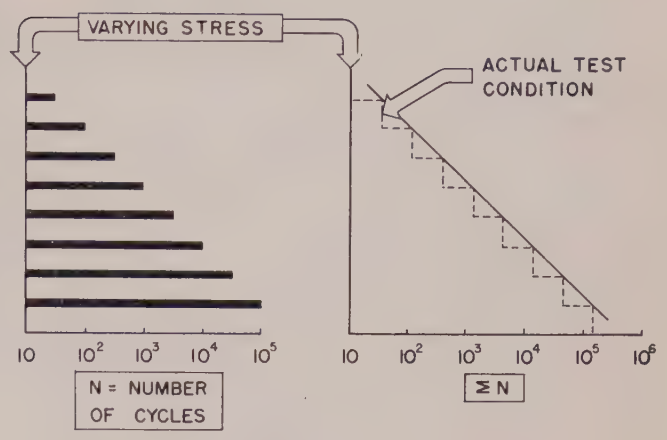

Fig. 9 – a. (Left): Basic loading spectrum for step testing. b. (Right): Ordinary method of plotting loading spectrum summation from Fig. 9a.

Those operating fighter aircraft must recognize that constantly repeated violent training maneuvers, such as high-angle air-to-ground gunnery pull-outs, use up the available fatigue life at an accelerated rate, and the life of this aircraft may be only 10% of another subjected to normal usage. The military should consider this fact when planning training programs.

With the above considerations in mind, a specified design gross area stress is used as a limitation

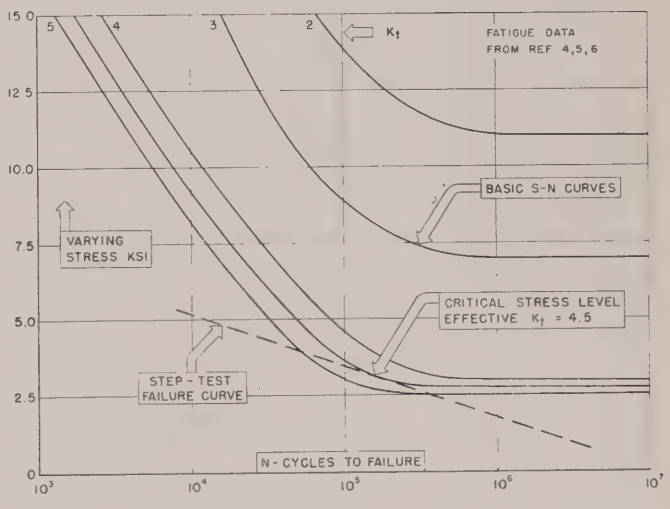

Fig. 10 – Method of estimating effective K_t from step test.

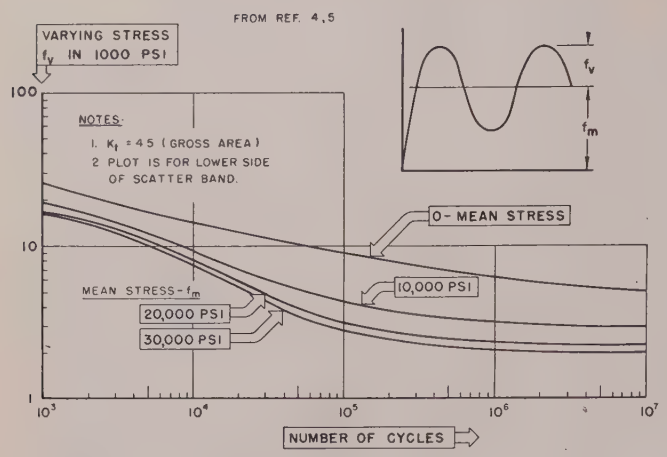

Fig. 11 – *S-N* curves for aluminum alloy with $K_t = 4.5$

in structural design. As soon as the design is "frozen," fatigue step tests should be run on wing lower surface tension joints and important discontinuities to verify the achievement of $K_t \leqq 4.0$ or whatever goal was set. This action will insure a design consistent with the starting assumptions.

Fatigue Analysis of Transports

If an accurate loading spectrum is known, then the fatigue analysis of transports is the same as for fighters. The trends of gust loading may be well established, but there remains a doubt concerning the accuracy of quantitative values. Considerable work remains to correlate these data with actual aircraft fatigue experiences before confidence can be established.

Lacking confidence in the quantitative values of published gust-loading data, the transport designer must turn to actual airline fatigue history. If he is unfortunate enough to have first-hand experience with such troubles, the job is made easier because he then has a fixed point or datum airplane to work from. The procedure he might use would be to calculate a fatigue-loading spectrum for the new model and for the old one. The same basic data would be used in each case with proper evaluation of speeds, altitudes, routes, and other items. The designer should have step-test fatigue data for the weak point of the datum airplane so that the value of K_t is known. Fig. 14 plots the life versus K_t for a typical transport and was derived from NACA gust data, *S-N* curves, and Miner's theory of cumulative damage.

If the engineer has constructed such a curve as Fig. 14 for his datum airplane, he can plot the actual failure point *A* and then predict what the life would be if K_t was changed to the new value *B*, set as the design goal for the new model. Next, he can construct Fig. 15 for the datum airplane and determine what the gross area stress would have to be in order to give the desired life. This is done by transferring point *B* of Fig. 14 to Fig. 15 and plotting point *C* at the desired life. These steps are accomplished by assuming the curves are of the proper shape but shifted to pass through the datum point.

Next, a curve similar to Fig. 15 should be constructed for the new model. Knowing that the curve was constructed from the same basic gust data as was for the datum airplane, the engineer will assume that the same errors of prediction exist in both. The necessary gross area stress to give the desired life can be obtained by plotting a point along a horizontal line through the desired life and in the same position relative to the calculated curve as was point *C* of Fig. 15.

There are many who have no first-hand experience with fatigue that can be used to establish datum points for such an analysis. An organization such as the NACA should gather and analyze these data and publish reports for the industry's use. It is important that the analysis be published in its complete form so that the engineer can reproduce the calculations for his airplane and make comparison on exactly the same basis.

Curves in Figs. 14 and 15 are of special interest as they emphasize two points:

(a) Life is very sensitive to K_t. Reducing K_t from 5 to 4.5 in this case increases the life by a factor of 6.

(b) Life is also sensitive to the design ultimate gross area stress. Reducing the stress from 60,000 to 50,000 psi increases the life by a factor of approximately 3.

Reducing K_t to the lowest practical limit is the first and most obvious thing to do. Further life improvement can then be obtained in only one manner and that is to design to a lower gross area stress. This simply means that you must add more area to the wing lower surface.

An interesting observation made during the fatigue analysis of a high-altitude transport is worth mentioning here. Airplanes that fly above 30,000 ft will be subjected to much less fatigue damage during the cruising period than one which cruises below 10,000 ft. It will be found that a large share of fatigue damage to the high-altitude airplane occurs during the climb and descent, and these portions of the flight must be given considerable attention. Further, it will be found that maneuvering loads play a more important part in the life of these aircraft and should not be ignored.

General Design Plan

The discussion of fighters and transports established that a permissible stress concentration factor should be chosen at the very start and that Lockheed experience has shown that $K_t = 4.5$ is the lowest safe value for design. Further, the discussion indicated how an allowable design gross

area stress could be determined that would provide the required service life for a structure having the chosen stress-concentration factor. In choosing the design gross area stress, the type of structure and consequence of crack propagation should influence the degree of conservatism. A fatigue crack in single- or two-spar construction cannot be discovered before it reduces the static strength to a dangerous level. This construction should either be avoided or the design stresses lowered so that any possibility of fatigue cracking is eliminated. If fatigue cracks are to occur, the best structure is one in which the crack could grow large enough to be easily detected before it would appreciably reduce the overall static strength. This structure probably consists of conventional skin and stringers with small spar caps and no blind areas to hide cracks.

The supervision of stress analysis during the design development stage can be relatively easy as only two decisions are required:

1. Design gross area tension stress. (In general this fixes the amount of structural material.)
2. Maximum permissible stress-concentration factor. (This fixes the detail design quality.)

The stress engineers then can proceed with the design with no thought of life prediction, and their only responsibility is one of good detail design and staying within the gross area stress level assigned.

Each joint and important discontinuity in the wing tension surface should be step tested in fatigue to verify the permissible stress-concentration factor.

If the tension-stress limitation for fatigue reasons will permit the use of weaker alloys without a weight penalty, then it would be best not to use 75S or 78S in the wing tension surface. Although these alloys are as good in fatigue as 24S and 14S, they are more expensive, brittle, susceptible to stress corrosion, and more difficult to handle in the factory.

It would be well to mention a few design pitfalls that should be avoided where possible.

1. Joints that cause visible flexing of stringers under repeated loads will usually be poor in fatigue. Such flexing may be caused by eccentric fitting attachments.
2. Bearing stresses should be kept low in rivet or screw attachments. High bearing stresses give unusually high stress-concentration factors at holes.
3. Design for consistent deflections. If stringers are spliced and skin is continuous, the skin may be overloaded because of joint flexibility.
4. Avoid the use of small stressed doors. Joint flexibility due to reasonable hole and screw tolerances cause the door to be ineffective at the small loads which cause fatigue, thereby overstressing adjoining structures.
5. For transports, buckling of shear webs should not be permitted below 1.3*g*. If shear webs are buckled during normal flight, flexing may cause early cracking.

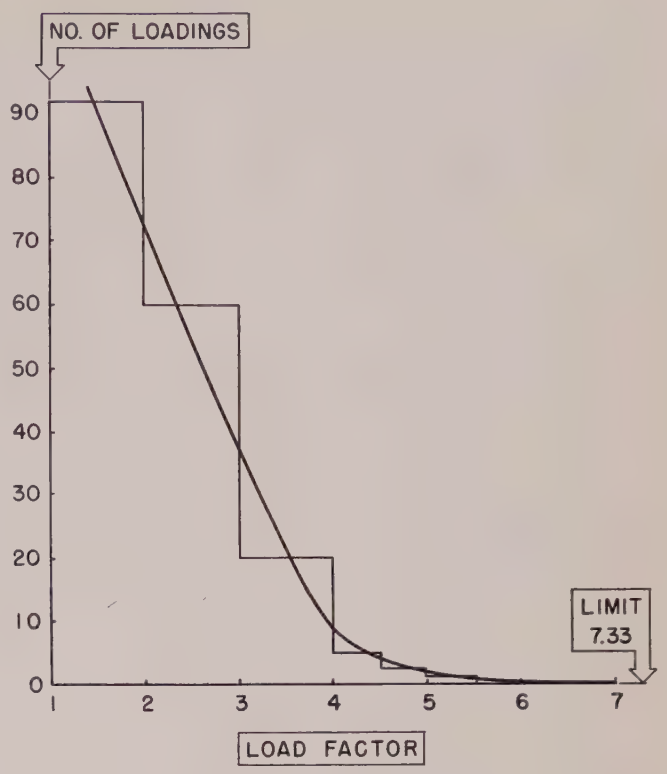

Fig. 12 – Loading history per flight hour assumed for fatigue-life analysis of a typical fighter

Maintenance

No matter how well designed, the fatigue life of any structure is controlled to a large extent by the

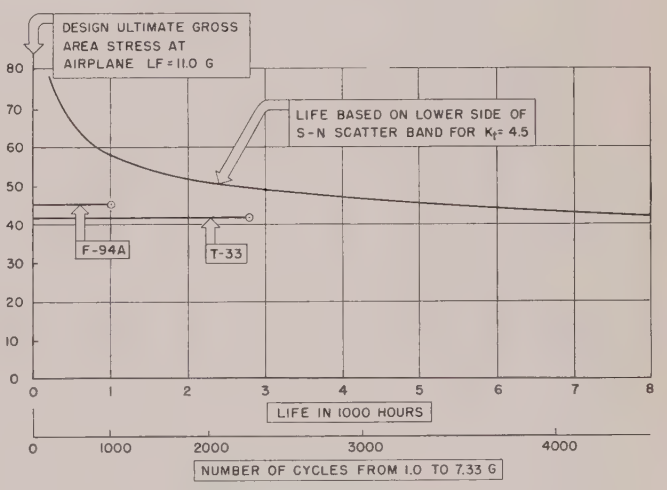

Fig. 13 – Prediction of life versus the design gross area stress

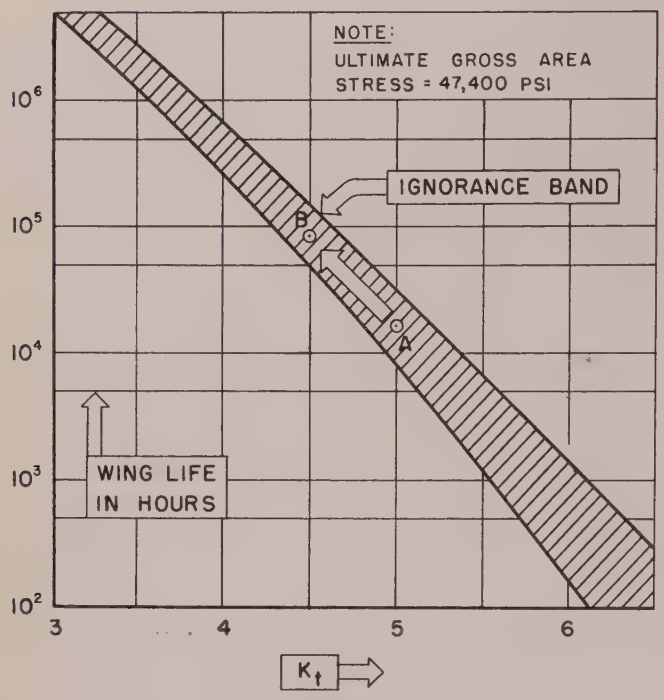

Fig. 14 – Wing life versus K_t for typical transports

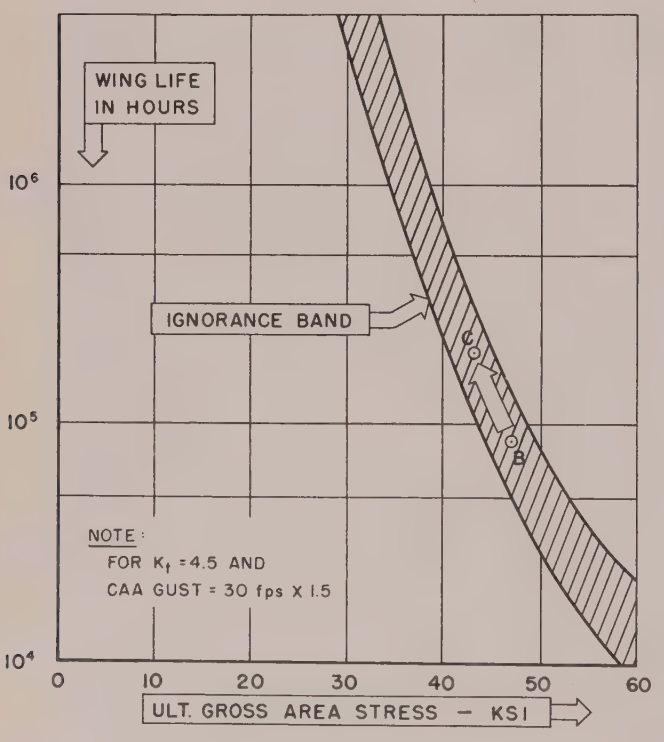

Fig. 15 – Wing life versus gross area stress

workmen, both in the factory and in the field. It is the mechanic that actually produces the geometry so necessary to insure fatigue life. A file mark or a carelessly dropped bucking bar can undo all that the designer has strived to accomplish. Workmanship, therefore, must be inspected for quality and conformance with the drawing and specifications. However, human errors do get by, and early isolated fatigue troubles are possible.

The modern automobile gives remarkably good service under ordinary conditions, but continuous use over rough dirt roads can break it down and wear it out in a short time. The airplane too can suffer from abuse. Turbulence and pilot technique over different routes will vary considerably; and, like the automobile, one operator may experience early fatigue troubles and another operator may never.

Inspection is the ultimate control over the fatigue problem. The alert operator will set up an inspection schedule designed to detect fatigue cracks before they become dangerous. As airplanes become older the operator must tighten inspection procedures and schedule them more frequently. By this method he can operate safely even though he has found and repaired many primary structural cracks. Fatigue will cause retirement of an airplane when inspection and maintenance costs make it uneconomical to operate, but the airplane should be outdated for other reasons before that time comes.

Inspection should systematically cover all vital joints and structural discontinuities of the wing lower surface and other tension loaded structure. The manufacturer can help in the preparation of such a checklist. Visual inspection should be made where possible with the aid of dye-penetrant methods in questionable spots. Portable X-ray equipment can be used for inspection of many blind areas and prevent costly and time-consuming tear-downs.

Summary

Today's aircraft are vulnerable to fatigue because the design stresses are higher than they were in the past and because they are expected to operate for a longer period of time. The *S-N* curves for newer aluminum alloys are no better or worse than the old ones, and if one expects more from the new materials they will give more fatigue problems.

The 1.0*g* operating stress level has frequently been used as a yardstick to compare fatigue life of similar aircraft. This is shown to have no direct bearing on fatigue life and should not be used. Instead, it is better to compare design ultimate gross area stresses.

The stress-concentration factor, K_t, has a powerful effect on fatigue life. Tests on many typical aircraft joints have shown there may be a lower

practical limit of $K_t = 4.5$ for design. To base the life of a conventional design on a lower value of K_t might be wishful thinking and have poor chance of success.

The peak stress within a conventional wing structure is shown to be almost a direct function of gross area stress and, within practical limits, not affected by net section stress if gross stress is held constant. For this reason, the gross area stress has been selected as the parameter for evaluating fatigue design problems. Should fatigue cracking occur, the best structure is one in which the crack can progress to a considerable extent and be found in routine inspection before causing a serious reduction in static strength. Concentrating most of the tension material in heavy spar caps is poor practice because a small crack is hard to detect and can cause a serious reduction in static strength. The best structure probably is of the conventional skin-stringer type, well distributed across the lower surface and with relatively small spar caps. Blind areas should be avoided so that visual inspection can detect fatigue cracks. Integral fuel tanks are advantageous in this respect as fuel leaks may give early warning of cracking.

Varying loads which cause fatigue come from flight maneuvers, atmospheric gusts, and ground loads. Fighters receive most of their fatigue damage from flight maneuvers. Transports accumulate most of theirs from atmospheric gusts. There can be a large scatter in fatigue damage received by a group of identical aircraft while operated under different conditions. This scatter may be as large as 100 to 1. The state of the art is not well established in this respect and the engineer must be on guard against overconservatism. It would be better not to penalize the majority of an aircraft type because a few might be subjected to abuse. Instead, the aircraft should be designed so that cracks could be detected before they became dangerous; and if the aircraft is abused, inspections should be more thorough and frequent.

Quality of design is measured by the effective elastic-stress-concentration factor, K_t. The state of the art is not yet developed to the point where one can always design a complicated joint and be certain that the desired quality standard, K_t, has been met. Fatigue testing of joints and important discontinuities appear necessary to verify design quality. Step testing to a typical loading spectrum is believed best as it eliminates many of the assumptions necessary to interpret the results from other types of tests.

Predicted fatigue life of a typical fighter versus ultimate design gross area stress is presented for an assumed loading spectrum and $K_t = 4.5$. It is shown that there is no direct relationship between life and constant amplitude testing of from $1.0g$ to $7.33g$. Also, the gross area stress must be limited to about 31,600 psi ultimate if the structure is expected to last for 10,000 cycles of from $1.0g$ to $7.33g$. Such an arbitrary requirement would produce excessive life and weight.

The life of a transport as well as a fighter is controlled by K_t and the gross area stress. Curves are shown for a typical transport to illustrate how sensitive life is to these two parameters. Further, it is shown how such curves can be used for life prediction of a new model based on the known life of one in current service.

Steps to be taken when designing for fatigue are listed as:

1. Selecting a type of structure that lends itself to easy fatigue-crack detection.
2. Setting a design quality standard (K_t).
3. Choosing a design gross area stress to provide the desired life.
4. Step testing to verify design quality.

Importance of the factory mechanic and inspector is stressed. Ultimate control over fatigue is the responsibility of maintenance crews. Inadvertent damage or even modifications may alter structural life. Maintenance inspection of primary lower surface structure should be scheduled at regular intervals, and vigilance should be increased as the airplane grows older. Blind areas should be given particular attention, and X-ray is a promising inspection method for such places.

Acknowledgment

The author is indebted to the Lockheed Structural Methods Department for much of the material presented in this paper. In particular, he wishes to express appreciation for the contributions of M. G. Childers, M. A. Melcon, J. E. Wignot, D. E. Nelson, R. L. McDougal, and R. A. McLellan who performed most of the calculations and prepared all the figures. Much of the background material and test data on which this paper is based was developed in the Lockheed Research Laboratory under the direct supervision of H. W. Foster.

DISCUSSION

Questions Author's Conclusions On Effects of Damaging Gusts

— E. W. Thrall
Douglas Aircraft Co., Inc.

THE author has presented some helpful suggestions for the problem of design for fatigue. The writer has been conducting fatigue tests to determine fatigue life of the aircraft structure and welcomes the new ideas suggested in the paper. It is granted that lg stress level might

not be a good basis for measuring relative fatigue lives but the statements made in the paragraph under "operating stress levels" are not convincing. The author here states that the airplane of high load factor is more susceptible to damaging gusts of 5 fps than is the airplane of low load factor and then in his conclusions states the opposite, namely, that the airplane of low load factor is critical for gusts and the airplane of high load factor is critical for maneuvers. The latter has been the writer's experience. Perhaps there is a little more information available to the author that bears out his statements.

The writer has studied various reports on "step" testing and has not found a usable procedure. Before the information presented by the author can be used it is necessary to have considerably more data than he has presented in the paper. First, the loading schedule used for transport- and fighter-type airplanes would be most desirable. Then it is necessary to know exactly how the data are transferred to an S-N curve such as is shown in Fig. 10. With this more detailed information it would be possible for other tests to be run to substantiate the data indicated in the subject report.

Says Load Factor-Time Data Needed for Fighter Operation

– S. B. Pfeiffer
Douglas Aircraft Co., Inc.

THE author mentions that statistical data on maneuvering loads are available from the Air Force and Navy. He also presents a maneuver load spectrum in Fig. 12. A comparison of Fig. 12 with other available data on maneuver loads shows that it falls roughly in the center of an extremely wide scatter band. It appears, then, to be as good an estimate as any for a "fighter" load spectrum. However, the scatter band of maneuver loads is composed of fragmentary information on various types of operation and the spread is so large that almost any desired load spectrum could be taken.

Fatigue from maneuver loads is already a problem and with the advent of XA78ST material the situation may become serious, especially if the present philosophy of setting the stress level on the basis of ultimate strength only continues. Consequently, the writer would like to suggest that the accumulation of considerable load factor-time (not V-G) data for various types of fighter operation is essential. With this the applicable load spectrum for each new design could be determined and adequate "design for fatigue" would be more possible.

Author's Closure To Discussion

The section of my paper concerning "operating stress levels" must not have been clear otherwise Mr. Thrall might not have made his first comment. The point being discussed was the comparison of two airplanes critically loaded by gusts that determined their ultimate design strengths. For example, the Constellation is a low load factor airplane and could be assumed critical for gusts. A jet transport might also be critical for gusts and, due to its high speed, be a high load factor airplane. If both these airplanes were designed to the same ultimate gross area stress and had the same quality of detail design and flew the same routes, distances, and altitudes, then the high load factor airplane could have a shorter life in miles flown. On the other hand, fighters are critically loaded by maneuvers and life is quite short in comparison with that of a transport because of the many violent maneuvers these aircraft are subjected to in training and service operations.

ORAL DISCUSSION

Reported by L. P. Spalding
North American Aviation, Inc.

Question: In the step testing system, are the various loads applied at random and what is the correlation between such applications and flight conditions?

Answer: The steps of load are applied in a systematic manner, that is, load steps are gradually increased and decreased rather than being applied in a random manner. The complete up and down sequence may be applied as many as 40 times before failure. It is believed that this system approximates random loading.

Comment: The lack of adequate gust data may not be as severe as indicated in the paper. NACA has collected gust data since 1930, although much of the data are at higher gust levels. Since 1947 time-history data on all transport gusts have been collected in an organized manner. Transport data have been collected to a 25,000-ft altitude and military data to a higher altitude. On the above basis the gust spectrum for transports can be predicted with reasonable accuracy.

Comment: As an airline operator we are especially interested in the problem of fatigue. In a new design airline structural engineers review with the manufacturer's stress men potentially critical joints and establish a manual including marked photographs for the guidance of the maintenance mechanics in periodic examination of joints. This provides good control at present, although blind joints do pose a problem. X-ray is unwieldy and difficult to incorporate in portable equipment.

Question: What photoelastic testing is currently being done in connection with the fatigue problem?

Answer: Very little photoelastic testing is being done, although much fatigue testing is being made.

Question: Do you predict wing life from curves?

Answer: S-N curves have a tremendous amount of scatter. Design of an airplane that will never fail in fatigue is impracticable, hence the design is based on finite life with reasonable usage.

Question: Are there different stress concentration factors for different hole sizes?

Answer: NACA data on specific hole sizes were used only as a yardstick for relative structural efficiency. Detailed consideration of such variations as might occur with different hole sizes has been bypassed in order that an overall yardstick might be secured.

Question: Has work been done on the problem of prestressing by peening or rolling to avoid high localized stress?

Answer: Much work has been done on prestressing, but only the overall effect on wing structure is being considered here. The problem of eliminating stress risers at every hole in a wing structure would be an enormous one.

Comment: I concur that advancement of the art is occurring. Past emphasis has been on commercial aircraft while military aircraft are now receiving considerable attention. Much of the work has been done on wings. Similar study should also be done on other portions such as the tail, continuing correlation of gust velocity with actual performance.

HYDRAULIC STEERING IN GENERAL MOTORS CARS

C. W. Lincoln, Saginaw Steering Gear Division, GMC

This paper was presented at a meeting of the St. Louis Section of the SAE, Feb. 5, 1953.

THE history of automobile steering in general is that of a continuous struggle to keep steering effort within reason, as the size and weight of cars have increased. The first move was to discard the vertical tiller shaft, with its direct linkage to the steering knuckles, in favor of a steering wheel which operated through a reduction gear (Fig. 1). As this move was made, the automobile steering gear, as we think of it today, was born.

Further development has proceeded along two lines. One method has consisted in merely increasing the ratio of the gears. Of course, there was a limit to progress in this direction. When we got to the point where we didn't have time for enough wheeling to get the car around a corner, then we had to quit. The other method has consisted of increasing mechanical efficiency within the gearbox, and here again there is a limit to progress. We obviously have to stop somewhere short of 100%.

The most recent type of steering gear is that in-

THE Saginaw design of hydraulic steering gear is described in this paper.

This gear employs a screw and nut, with a novel system of circulating balls between these two members. The motion of the nut is transmitted to the cross-shaft or pitman shaft by means of rack and sector teeth.

Advantages claimed for this design include simplicity and versatility. The loads of the centering springs and even the size and number of reaction plungers can be varied, making the steering "firm" or "soft," as desired.

The Author

C. W. LINCOLN (M '47) is chief engineer of the Saginaw Steering Gear Division of General Motors Corp. He received his B.S. degree in mechanical engineering from the University of Illinois. After a period of enlisted service during World War I, Mr. Lincoln was employed by the Carroll Engineering Co. of Dayton, Ohio and the Lufkin Rule Co. of Saginaw, Mich. Mr. Lincoln joined the Steering Gear Division in 1932 and since that time has served in various capacities in the engineering department.

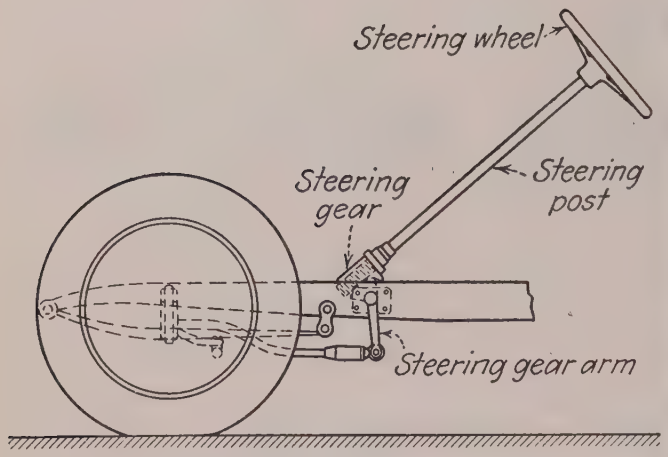

Fig. 1 – Steering wheel operated through reduction gear

Fig. 3 – Earthmoving equipment became so heavy that some form of power steering was needed

troduced by Saginaw in 1941. This gear employs a screw and nut, with a novel system of circulating balls between these two members (Fig. 2). The motion of the nut is transmitted to the cross-shaft or pitman shaft by means of rack and sector teeth.

During the later years of development of manual steering gears, some vehicles, particularly those designed for road construction and maintenance, and for earthmoving generally (Fig. 3), were already becoming so heavy that manual steering was approaching the impossible. The only out was the addition of power to the efforts of the driver at the steering wheel. This situation was actually reached about twenty years ago, so that power steering, in itself, is no new thing.

From the beginning, we have had in practically

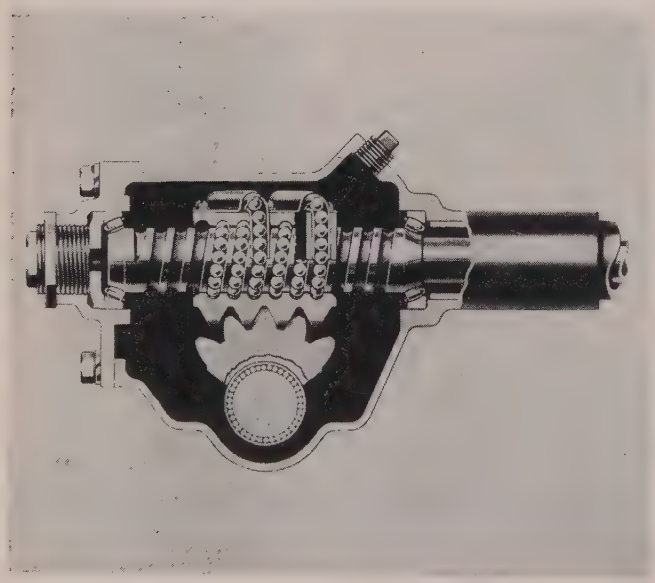

Fig. 2 – Cutaway view of Saginaw steering gear showing system of circulating balls

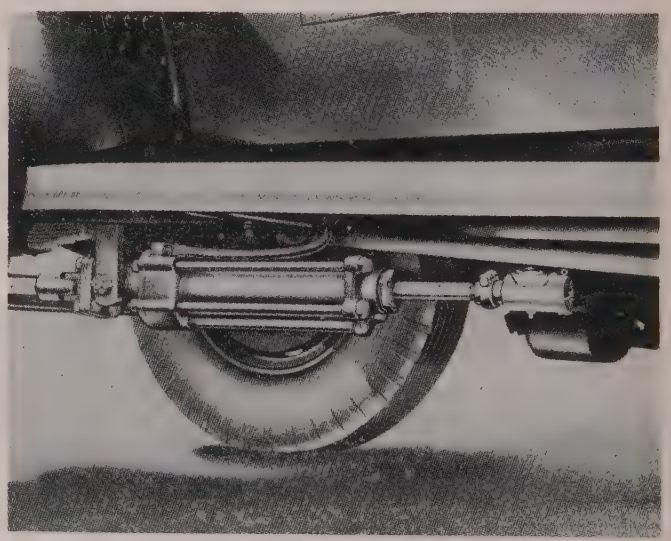

Fig. 4 – Early commercial installation was called a linkage booster

all cases a fluid pump driven by the engine, a power cylinder for converting fluid pressure into mechanical push or pull, and a control valve operating automatically with the turning of the steering wheel. With few exceptions the fluid has been oil. The earliest commercial installations were what we call linkage boosters (Fig. 4). In these the manual steering gear is left substantially untouched, the control valve is installed in the linkage between the steering gear and the vehicle wheels, and the power of the cylinder is applied to the linkage. Later, what we call integral hydraulic gears (Fig. 5) came into the picture for heavy vehicles. In these the control valve is actuated by the motion of the steering shaft, the hydraulic cylinder is mounted on the steering gear housing, and the power is applied to the cross-shaft of the gear.

Saginaw began its work on power steering twenty years or more ago, went into production on integral hydraulic gears for armored vehicles at the outset of World War II, and has for about five years been in steady production on similar gears for off-the-road vehicles for Caterpillar Tractor.

As all of us know, several of the passenger cars have now followed the lead of the heavy vehicles, and are currently offering hydraulic steering. At present, cars of Chrysler manufacture can be purchased with an integral hydraulic gear made by Gemmer, and Cadillacs, Buicks, and Oldsmobiles with integral gears made by Saginaw. The Gemmer gear has been described in a paper by W. A. Hunter of Chrysler.[1]

The Saginaw hydraulic gear for passenger cars is, in effect, simply a scaled-down version of the gear that has been in production for heavier vehicles during and since the war. The basis of the gear is the well-known circulating ball gear (Fig. 2), as designed for purely manual steering.

In the power gear (Fig. 6), a double-acting hydraulic cylinder is bolted to one end of the steering gear housing, with the free end of the piston rod projecting into the housing. The pitman shaft gear, which in the manual gear is provided with a single set of gear teeth meshing with the rack of the ball nut, is in the power gear provided with an additional set of teeth opposite the first. A power rack pinned to the end of the piston rod meshes with this additional set of teeth.

In the double-acting hydraulic cylinder and the power gearing, means are provided for rotating the pitman shaft by manual effort through the ball nut, by power obtained from hydraulic pressure against the piston, or by a combination of the two, provided, of course, that there is a source for the hydraulic pressure, and a controlling device that will direct

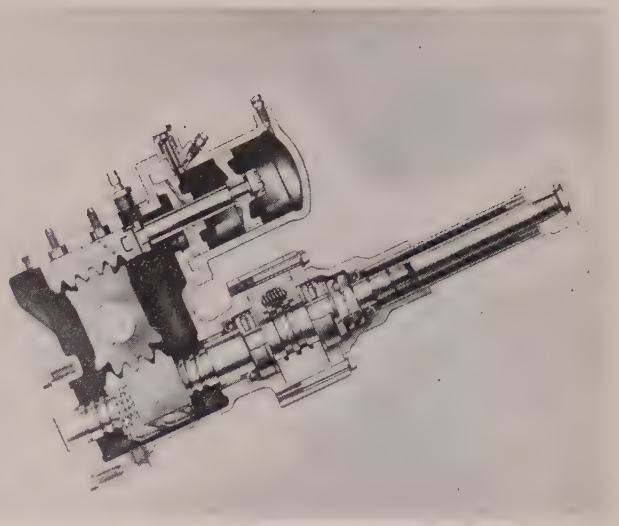

Fig. 5 – Integral hydraulic gear used on heavy vehicles

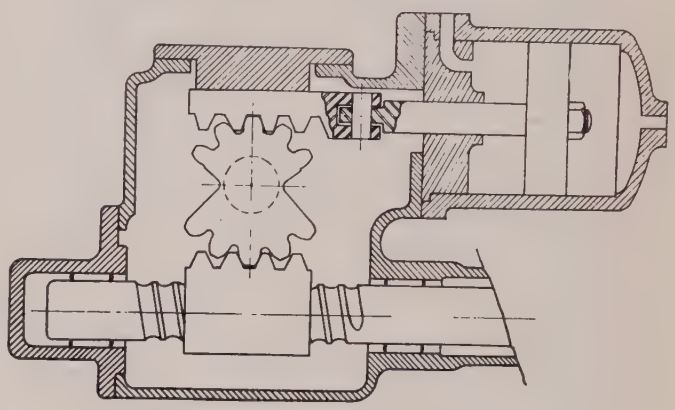

Fig. 6 – Power gear showing double-acting hydraulic cylinder bolted to steering gear housing

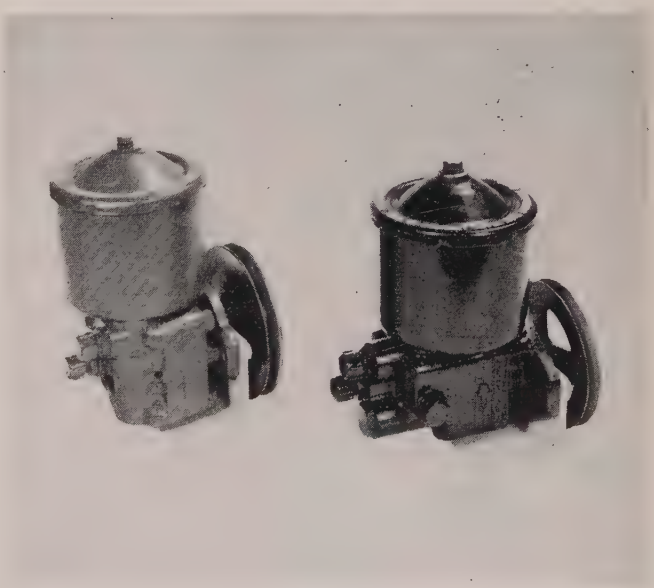

Fig. 7 – Hydraulic pumps

[1] See "Hydraulic Power Steering," by W. A. Hunter in Proceedings of National Conference on Industrial Hydraulics. Vol. 5, 1952, Armor Research Foundation, Chicago.

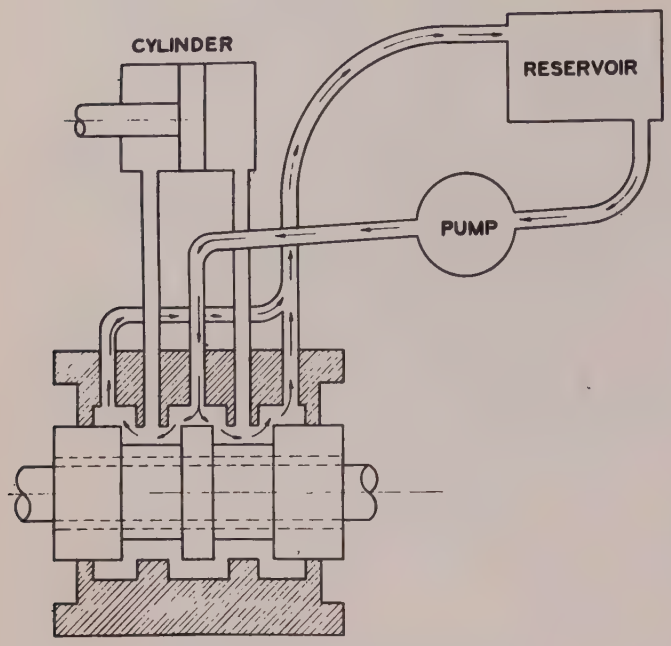

Fig. 8 – Control valve

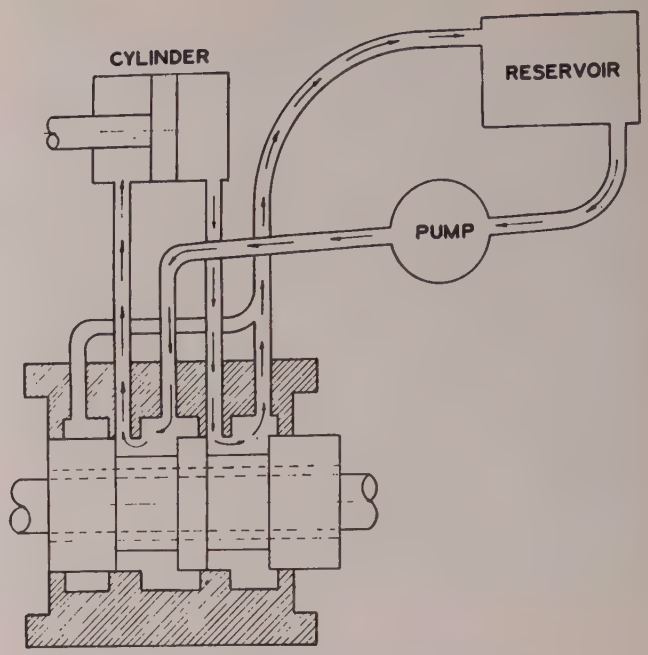

Fig. 9 – Control valve with spool displaced from centered position

the pressure to one end of the cylinder or the other when it is wanted.

The source of pressure is a hydraulic pump of either the vane or the rotor type (Fig. 7). The pump is mounted on the engine of the car, and is driven by V belting from a pulley on the crankshaft. A small oil reservoir is mounted on the pump, and communicates directly with the pump intake.

For the hydraulic controlling device, a single sliding valve is used. In this a round inner member, or spool, slides with a close hydraulic fit within a stationary outer member, or valve housing (Fig. 8). The spool is hollow, permitting the steering shaft to pass directly through the valve. The valve housing is provided with three annular grooves in the bore, and the valve spool with two annular grooves in its exterior surface, and there are holes through the wall of the valve housing by which these several annular grooves communicate with the exterior, where connections are made to the discharge side of the pump, the return to the reservoir, and the two ends of the power cylinder.

The annular grooves in the bore of the valve housing are spaced somewhat apart. The center groove is connected with the discharge orifice of the hydraulic pump by means of a flexible hose, and the two side grooves communicate with a second flexible hose, which carries return oil to the reservoir. When the valve spool is centered, each one of the two grooves in its exterior slightly overlaps the corresponding land in the bore of the valve housing. Thus, oil can flow not only into each one of the spool grooves, but also out of each one into the corresponding return groove of the housing.

The holes through the wall of the valve housing which communicate with the two annular grooves of the spool are connected, by means of steel tubing, one to each end of the power cylinder. When the valve is centered, there is, of course, no differential in pressure between the two grooves of the valve spool, and consequently no tendency on the part of the hydraulic system to move the piston either one way or the other. Furthermore, if the piston is moved otherwise than by the hydraulic system, as by manual operation, for example, oil is free to flow from one end of the cylinder to the corresponding valve spool groove, thence to the lower end of the return line through the return groove of the valve housing, and finally through the other return groove and the other spool groove back into the opposite end of the cylinder. The centered, or open, valve is a sort of free-for-all passage for oil from the pump to the reservoir, and for oil from either end of the cylinder to the other. With the valve in this position, the steering is purely manual.

Now, if the valve spool is displaced from the centered position in one direction (Fig. 9), then one of the grooves of the valve spool will be moved completely out of communication with the pressure groove of the valve housing, so that oil from the pump can flow into the other spool groove only. At the same time, this other spool groove has been moved completely out of communication with the return groove of the housing, so that the oil cannot go anywhere from the spool groove except out through the radial hole in the valve body to one end

of the cylinder. While this is going on, the first spool groove, which has been closed to the pressure groove, is opened wider than ever to the return groove, so that oil from the opposite end of the cylinder is free to pass through the second radial hole in the valve housing, through the spool groove in question to the return groove, and finally through the return line and the reservoir back to the pump.

With the valve spool displaced in this manner, there is full hydraulic action, with the pump exerting all of its pressure against one side of the piston. If the valve spool is displaced from center in the opposite direction, then the pressure of the pump will be against the opposite side of the piston.

Now, to return to the manually operated circulating ball gear (Fig. 2) for a moment, in that gear the screw, or worm, is mounted in "spherangular" roller bearings, which confine the worm endwise as well as radially. Thus, the only motion the worm can have is that of rotation. In adapting the gear to the use of hydraulic power, however, the "spherangular" bearings are replaced by needle bearings (Fig. 10). These latter confine the worm radially, but allow endwise movement. To take the thrust loads on the worm, ball thrust bearings are provided in addition to the needle bearings. These thrust bearings are disposed one above and one below the valve spool, and the "string" of three parts, lower bearing, spool, and upper bearing, is pulled up against a shoulder of the worm by means of a nut above the upper bearing. The races of the ball thrust bearings are larger in diameter than the valve spool but they are small enough so that they can move freely into counterbores provided in the two end faces of the valve housing. The distance between the bottoms of these two counterbores is somewhat less than the length of the valve spool. With this construction the bearings system allows

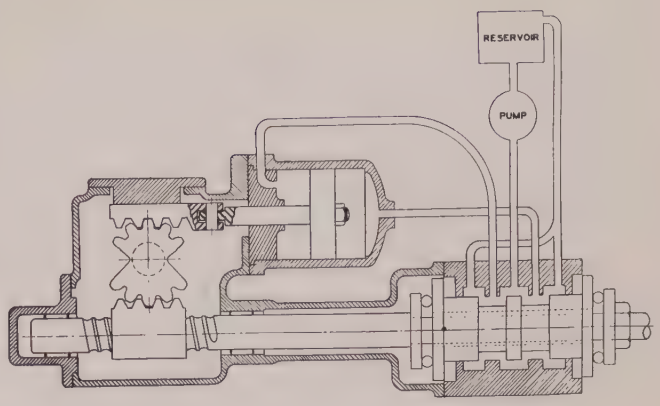

Fig. 10 – Power gear with needle bearings

a certain amount of axial play in that element of the steering gear which includes the worm, the ball thrust bearings, the valve spool, and the steering shaft, which is welded to, and substantially integral with, the worm.

It will be understood that when the valve spool is at the midpoint of the axial play referred to, the valve is in centered, or open, position, with no hydraulic assist to steering.

On the other hand, when the valve spool is at one extreme or the other of the axial play (Fig. 11), we have full hydraulic assist in one direction or the other.

To maintain the valve in the open position under normal circumstances, as in straight ahead driving, centering springs are added (Fig. 12).

The valve housing is provided with a number of lengthwise holes spaced in a circle about the bore. These holes are at a sufficient radial distance from

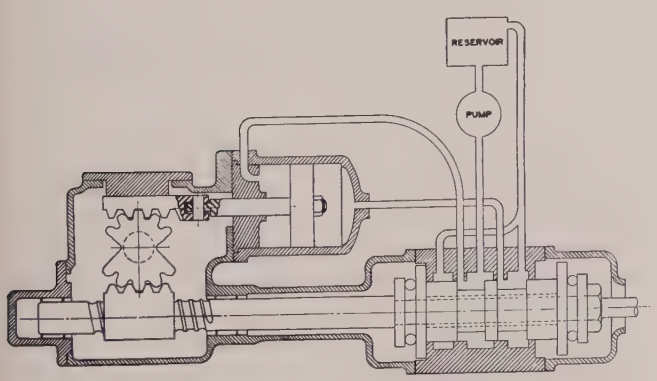

Fig. 11 – Valve spool at one extreme of axial play

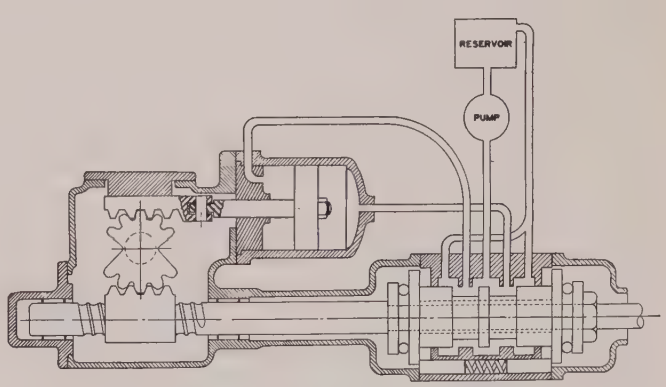

Fig. 12 – Valve held in open position by centering springs

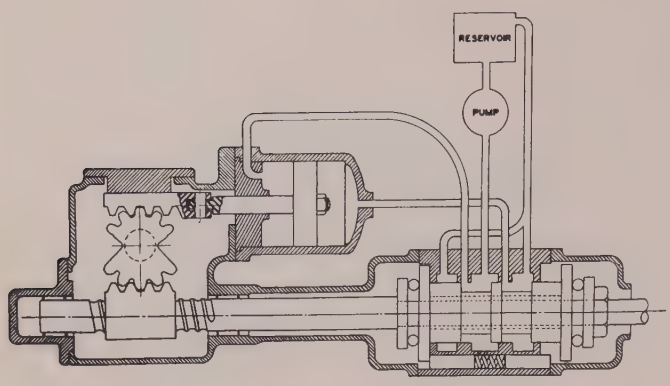

Fig. 13 – Centering springs compressed

the worm axis so that they avoid the annular grooves in the valve housing, and they are located so that they "dodge" the radial holes in the valve housing, which connect the various annular grooves with the exterior. Each one of the lengthwise holes is fitted with two plungers separated by a compression spring. The plungers are restrained from moving beyond the end faces of the valve housing, by shoulders of the steering gear housing and of a valve cover which is bolted to the upper face of the valve housing. Lengths of the various parts are so chosen that the springs are under a certain amount of initial compression when the device is assembled.

The ends of the spring plungers are covered partly by the shoulders of the fixed castings and partly by the races of the ball thrust bearings. At assembly the two bearings are actually drawn together by the nut on the upper end of the worm until the races of the bearings just pick up the spring plungers; so that the spring load on the plungers is divided between the stationary shoulders of the outer castings and the thrust bearings. In this situation the valve spool is just free between the two thrust bearings, and exactly centered in the valve housing. Thus, the spool is free of any tendency to bind within the valve housing, yet it moves endwise instantly upon any endwise movement of the worm and shaft.

The worm of the steering gear has a right-hand thread. With this thread, turning the steering wheel to the left tends to do two things. It tends to push the ball nut downward, and to push the worm, valve spool, and steering shaft upward. Which thing will happen depends, of course, on relative resistance to moving. The nut is in direct engagement with the pitman shaft gear, which has the pitman arm firmly affixed to its outer end, and the arm is attached to the steering knuckles of the front wheels by means of the front end linkage. Thus, any motion of the nut is a motion of the front wheels, and the front wheels always resist being turned. Their resistance may be great, as when they are far from straight ahead during standstill parking, or it may be slight, as in the case of a minor deviation from the straight-ahead position while driving at moderate speed. If the resistance of the front wheels to turning is slight, then the force which the worm has to exert on the ball nut will be slight, and the nut will move before the valve centering springs will compress. In this case, the valve will remain in its open position, and since the worm is, for the moment, anchored by the centering springs in a fixed axial position, the action will not differ from that of the former manual gear with the "spherangular" worm bearings. If, however, conditions are such that, as we keep on rotating the steering shaft, it becomes gradually more difficult to turn the front wheels, then the force which the worm must exert on the nut will keep increasing, until the point is reached where this force is just equal to the load on the centering springs. As the effort required to turn the front wheels increases beyond this point, the springs begin to compress (Fig. 13). As they do so, the valve spool shifts out of its centered position, oil is directed definitely against one side of the piston, and hydraulic effort is added to the manual effort of the driver to move the front wheels as required.

It will be noted that the hydraulic power system will turn the front wheels only by the angle corresponding to the angle through which the steering wheel is turned. Assuming that the driver had turned the wheel to the left, meeting sufficient resistance to close the valve and bring hydraulic assistance into action, the nut would continue to move downward as long as the driver kept turning the wheel. If the driver were to stop turning the wheel, the hydraulic pressure would momentarily continue to move the nut downward but in the absence of any rotation of the worm the nut would carry the worm, valve spool, and steering shaft downward with it, the valve spool would reach its centered position, switching the pump discharge back to free return to the reservoir, and steering would stop. For a right turn, the action would, of course, be the same, except that the motion of the nut would be upward.

The device of a slight axial movement of the worm, valve spool, and steering shaft to operate the valve results in an extremely simple mechanism, with a minimum of working parts. The use of a basic steering gear of the worm and ball nut type makes the employment of such a device a "natural," as the interaction between the worm and the nut remains exactly the same, regardless of the axial position of the worm.

Since the gear is, in effect, a purely manual gear within the range of slight resistance to steering, the effort which the driver must apply to the steer-

ing wheel within this range is, of course, directly proportional to the resistance. In short, within this range, the "road feel" is the same as in manual steering.

As the effort increases to the point where the valve spool begins to move, the valve ports begin to close. As they do so, changes in the fluid resistances of the various oil paths through the valve take place. This results in a differential of hydraulic pressure against the two sides of the piston, which keeps increasing until full closure of the ports is reached. Since each thousandth of an inch of movement of the valve spool means that much more compression of the centering springs, and since the springs have a definite rate, the steering effort itself is building up during this process. This results in "road feel" during closing of the ports, and a smooth transition from manual steering to hydraulic.

To continue "road feel" during full hydraulic steering, the construction at the pressure groove of the valve housing is somewhat modified. The major diameter of the groove has actually been increased, so that the groove communicates with the plunger holes in the region of the springs (Fig. 14). Thus, the inner ends of the plungers are exposed to the hydraulic pressure, and, as this builds up to overcome increasing resistance of front wheels to turning, the force tending to center the valve is increased. That is, the total force tending to center the valve consists of the load on the centering springs plus the load on the displaced plungers due to increasing hydraulic pressure against their inner ends. In this way the effort to move the steering wheel always increases with increasing resistance at the front wheels, regardless of whether the steering is in the manual range or the hydraulic range, and the driver has "road feel" for all phases of steering.

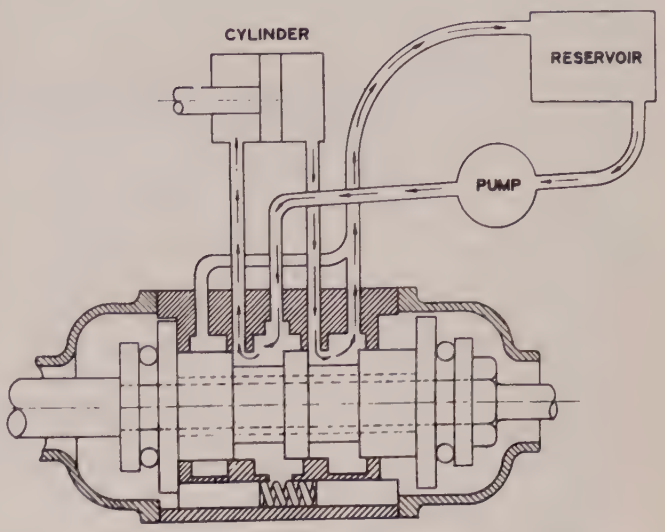

Fig. 14 – Pressure groove of valve housing modified to achieve "road feel" during full hydraulic steering

Reduction in Steering Effort

The reduction in steering effort is, of course, the greatest advantage of the gear. In fact, it is the gear's reason for being. This advantage is most appreciated in situations where the effort with manual steering is highest, and these are in parking and in turning city corners. The effort required with manual gears for road driving at road speeds is not great, and would not in itself warrant the additional cost of hydraulic steering. Actually, the slight efforts required in high-speed driving down a straight road are not less in the Saginaw hydraulic gear than they would be with a manual gear of the circulating ball type, as it was felt best to make no change whatever in the feel of steering under this condition. These efforts range up to about 3 lb of tangential pull on the rim of an 18-in. steering wheel. Consequently, the valve centering springs are so selected that they do not begin to compress until the wheel pull reaches 3 lb. The rate of the springs is so selected that, while the wheel pull is increasing from 3 lb to about 4 lb, the valve is shifting from the fully open position to the fully closed, and the transition from manual steering to hydraulic is taking place.

If further steering, at increasing resistance to turning, is required, the hydraulic pump does the bulk of the work, by building up the needed pressure in the cylinder. During this phase, the steering effort is increasing, but only because of the hy-

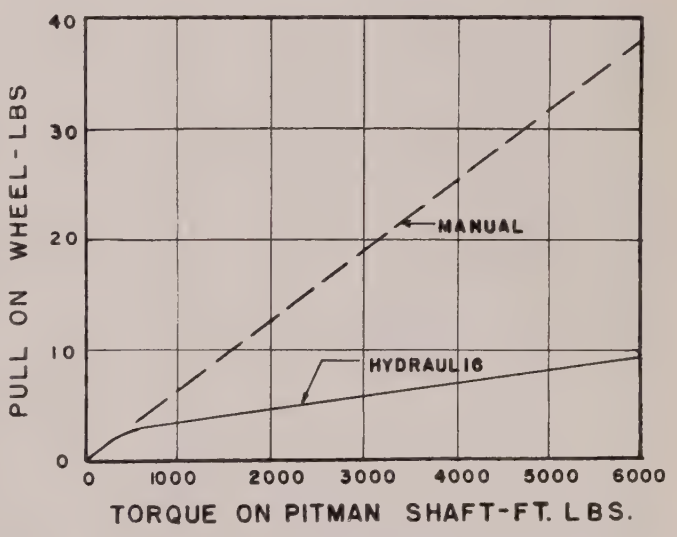

Fig. 15 – Pull on rim of steering wheel plotted against torque at hub of pitman arm

draulic reaction in the valve, and the increase in effort is far less than with a manual gear. Thus, a graph (Fig. 15) of pounds pull on the rim of the steering wheel plotted against torque at the hub of the pitman arm shows the effort identical with that of a manual gear up to the 3-lb wheel pull, with the steering effort for the hydraulic gear falling away from that of the manual during the transition period from 3 to 4 lb of pull. Beyond the transition period the effort for the hydraulic gear builds up slowly, so that at the maximum torque requirement, namely, in standstill parking on dry pavement, the driver is exerting a pull of only 8 to 10 lb on the wheel. With a good manual gear, the wheel pull that the driver must exert in a like situation ranges from 30 to 40 lb. The hydraulic gear provides greatly reduced effort for rounding sharp curves and city corners, and cuts the parking effort to about one-fourth of the old requirement.

Dampening of Road Shocks

While the reduction in steering effort is the greatest advantage of the hydraulic gear, the dampening of road shocks is a further and important advantage. The designers of high-speed automotive vehicles build into the front ends an inherent tendency to maintain the front wheels in the straight-ahead position, so that the vehicles will "track" while on the road, and will by themselves "come out of" a sharp turn. It is obvious that an irreversible steering gear would completely frustrate this tendency to "track." A steering gear must be a reversible mechanism. Some road surfaces are irregular, and the irregularities exert impacts on the front wheels tending to deflect them momentarily from their desired position. Since the steering gear is reversible, these impacts come through the gear back to the driver in the form of "jerks" on the steering wheel. While the driver with a manual gear may not be greatly troubled by impacts and "jerks" that are minor, all such disturbances contribute to driving fatigue.

With the hydraulic gear, however, impacts of any consequence immediately compress the valve centering springs, closing the valve momentarily, and imposing against the piston a solid block of incompressible hydraulic fluid. This, of course, forestalls the sudden tendency to rotate the wheel that would be felt in a manual gear. Experimental cars equipped with hydraulic gears have been driven through rutted roads, and off of and back onto pavement shoulders at road speed. The performance of the gear in dampening road shock under conditions of this kind is little short of spectacular.

While the prospect of a failure in the hydraulic system is remote, it was felt that car owners, with their natural suspicion of the complete success of any new device, might wish to be assured that there would be no impairment of control in the case of such a failure. For this reason, the gear has been designed in such a way that it reverts automatically to a normal manual gear in case of any hydraulic failure.

The ratio of the hydraulic gears now in production is 21.3. This ratio is exactly the same as that of the manual gear for the Cadillac car, for example, and it varies but little from that of any manual gear which the hydraulic gear is designed to replace. It was felt that drivers would have less to learn in handling cars equipped with the new gear, and would handle the cars more safely, if the amount of wheeling to be done in rounding a given turn was left substantially the same as before, the only difference being the reduction in effort. A good many years have been spent in accustoming drivers gradually to the present high steering ratios, and it may be smart to take at least a few years in reversing the process.

The retention of the present ratio works in well with reversion to manual steering in case of hydraulic failure. It means that in such a case the handling of the car would be no more difficult than it was before the advent of the hydraulic gear.

The simplicity of the Saginaw design of hydraulic gear has already been mentioned. Another advantage of the design, which may be mentioned in closing, is its versatility (Fig. 16). The loads of the centering springs and even the size and number of the reaction plungers can be varied, making the steering "firm" or "soft," as desired. Furthermore, automobile designers can take advantage of a wide variety of arrangements of the basic elements of the gear in order to get it into available space. As a matter of fact, the arrangement is somewhat different for each one of the three cars now fitted with the gear.

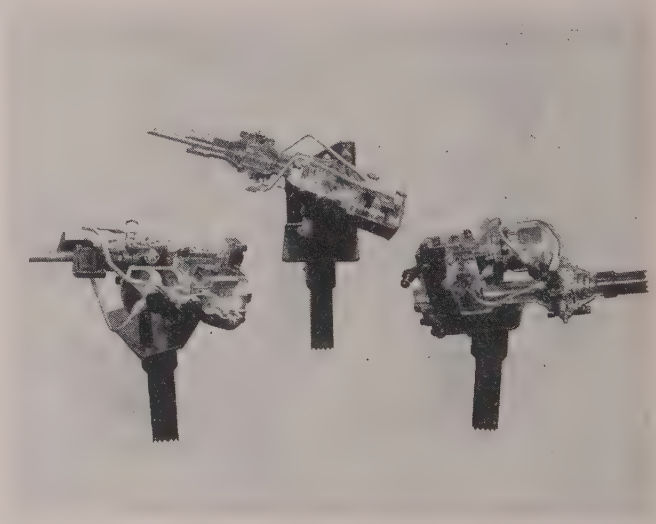

Fig. 16 – Three arrangements of Saginaw design

Deaerating Cold-Weather Oil Systems

F. E. Carroll, Jr., United Aircraft Products, Inc.

This paper was presented at the SAE Summer Meeting, Atlantic City, N. J., June 12, 1953.

RECIPROCATING engines are designed to be lubricated with air-free oil in the normal fluid state. Yet since the inception of the use of the piston engine in military vehicles, particularly aircraft, operational requirements have constantly plagued the engine designer with special oil-system conditions other than those for which the engine was originally designed.

The sequence of oil-system changes which have transpired is as follows: The first engines carried the oil supply in the crankcase – wet-sump engines as shown in Fig. 1A. With the advent of higher-horsepower engines with attendant increases in the oil-consumption rate, coupled with the range requirement, the oil supply could no longer be carried in the aircraft engine crankcase. The "dry-sump" engine then appeared with a separate oil-supply tank as in Fig. 1B. In the late '30's, early dry-sump-engine-aircraft crashes due to insufficient engine-scavenge-system capacity were numerous – oil foaming and spewing caused the failures. The scavenge-system capacity was then increased first 100% then 200% over the original scavenge capacity, and at the same time *over the pressure-pump* capacity (Figs. 1C and 1D). The result? The scavenge-pump excess capacity pumps air into the scavenged oil returned to the tank. Since the tank in effect was a large settling chamber through which the oil passed, the entrained air escaped from the oil sufficiently not to cause engine-oil-system difficulty. This satisfactory condition was short lived, however, inasmuch as a military requirement that ground vehicles and aircraft must operate at arctic temperatures was established.

It was then found necessary to complicate further the oil system by diluting the viscous oil with fuel in order to obtain sufficient fluidity for engine crankability after exposure to cold temperatures (Fig. 1D). But by diluting the entire tank oil supply, the usable oil available was considerably decreased (approximately 25%) after the fuel boiled off because of engine heat. To retain the same oil capacity but yet prevent loss of usable oil due to dilution of the entire supply, a short-circuit pipe or hopper was inserted in the tank between the inlet and outlet ports through which the oil passed (Fig. 1E). Immediately, the settling chamber for deaeration was eliminated and the aerated oil returned to the tank continued back to the engine – thus an oil-aeration problem was produced.

Now with the introduction of the hopper as part of the diluted-oil circulatory system, after engine shut-down *segregation* of the diluted circulatory system oil from the undiluted make-up oil surrounding the hopper became a major problem. The

AERATION and cold-temperature starting problems faced by aircraft-engine designers are said to be solved by use of the deaerating cold-weather oil system described in this paper.

In this system, all the oil in the tank circulates through the engine rather than the small volume of oil circulated by the conventional system.

The Author

F. E. CARROLL, JR., (M '47) is chief engineer of United Aircraft Products, Inc. He attended the Carnegie Institute of Technology. In 1942 he entered the Air Force and was assigned to the Power Plant Laboratory at Wright-Paterson Air Force Base as a project engineer on engine lubrication systems. Mr. Carroll joined United as assistant chief engineer in 1951.

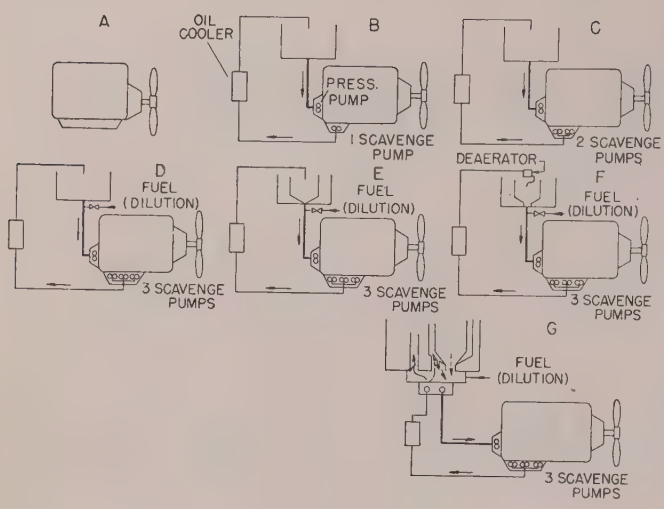

Fig. 1 – Oil-system evolution

first tank hopper design used an open passageway at the tank bottom as a connecting passageway for the make-up oil (Fig. 2A). This passage was later changed to a series of holes near the bottom on the periphery of the hopper (Fig. 2B). However, the undiluted oil seeped into the diluted hopper oil. To obtain positive segregation, the obvious method of providing gravity- or lightly spring-loaded check valves over the interconnecting ports was tried as in Fig. 2C. However, with check valves (after engine shutdown) the undiluted oil still persisted in seeping into the hopper, settling in the tank out oil

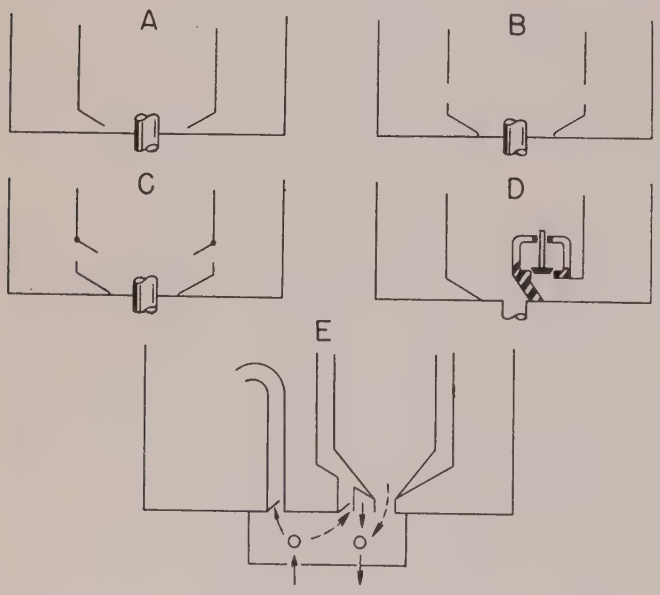

Fig. 2 – Oil-tank evolution

line and subsequently freezing during aircraft exposure at cold temperatures. An effective plug then was present in the engine supply line. On engine start, unless the pilot was quick to act, the engine was lost. A further development was to replace the flapper check valve with a lightly spring- or gravity-loaded poppet valve (Fig. 2D). However, the problem of undiluted oil seeping into the hopper and tank out line persisted.

Since all attempts to obtain positive segregation were fruitless, the obvious solution was to melt the frozen oil slugs by the use of brute force – the application of heat (large amounts of heat) for hours to the oil system before attempting an engine start. Such has been the cold-weather operational procedure. Several units of this external ground heating equipment of 400,000 Btu per hr capacity per unit is required *for the oil system* for from 1 to 3 hr (depending on ambient cold temperature and the specific oil system design) before an attempted engine start can be made.

The number of papers which have been written concerning the engine-oil-supply-aeration problem is staggering.[1] Even if one has never personally been involved in engine-oil-supply-aeration problems, after one reads the many referenced documents, it is apparent that a serious problem has existed and still does exist. It is logical to state that oil aeration is exacting a toll in terms of engine life. With regard to obtaining a satisfactory cold-weather oil system, the elimination of the time required for ground preheat would save equipment dollars of a large magnitude; but most important, from the time it is known that a flight is necessary, the elimination of ground heat for the oil system would enable arctic aircraft to be airborne in minutes – not hours.

Further, regarding cold-weather oil systems, the reader is referred to SAE Aeronautical Information Report No. 13A,[2] in which it is stated, "Although other forms of aircraft propulsion engines are being developed which may gradually displace the conventional engine, the development of cold-weather starting equipment should continue in order that military aircraft using conventional engines will have a quick and positive method of starting under cold-weather conditions."

This paper presents a fully developed oil system design which reduces oil aeration to an insignificant level and eliminates the cold-weather oil-system-starting problem. See Figs. 1G and 2E.

Oil Aeration

It will not be necessary to enter into a detailed discussion of an engine lubrication system or of various definitions for entrained air, foaming oil, and similar terms, because the referenced papers

[1] See references 1-8 under Additional References at end of this paper and also references given in footnotes 3, 4, 5, 6, and 8 of this paper.
[2] See SAE Aeronautical Information Report No. 13A, October, 1951, "Final Report on Cold Starting of Reciprocating Aircraft Engines."

have thoroughly covered these subjects. However, as described by these many referenced papers and by the above brief paragraphs, it is well known that the standard excess engine-scavenge-system capacity of from 200 to 300% over the pressure-pump capacity results in oil aeration of from 15% to 25% (by volume) returned to the tank through the hopper and thence to the engine. This aeration introduces a problem. R. J. S. Pigott, in his report "Oil Aeration,"[3] points up the problem based on extensive tests.

Briefly, he explains that obviously if a mixture taken into the suction side of a pump is 90% oil and 10% air (by volume), the pump cannot deliver over 90% of its normal capacity. But, he continues, it may not be so obvious why the delivery should in general be less than 90% of normal. Extensive tests conducted at the Gulf Research & Development Co. Laboratory pointed out the reasons. Pipe friction, pump internal loss, clearance volume of the pump, and pump slippage are listed as reasons for the additional loss. As an example of the effect of oil aeration on the engine, he points out that at a tank absolute pressure of 7.6 in. of Hg with 13.6% air by volume (in a typical spur-gear-pump oil system), the actual volumetric efficiency dropped from 95 to 63%. For the same conditions at ground level, the pump volumetric efficiency with 13.6% air at the tank was reduced from 95 to 78%.

Note that engine tank oil aeration conservatively is of the order of 15% at altitudes below 5000 ft. Further, it is generally agreed that entrained air (bubbles) cause continual oxidation of the oil resulting in the formation of sludge.

To eliminate this effect on the engine, including the lubricating pressure pump, it is obvious that air bubbles should be removed from the oil. The problem of bubble removal is controlled by Stokes Law if the flow is viscous; Rittingers formula applies for turbulent conditions:

Stokes Law:

$$v = \frac{2gd^2}{36}\left(\frac{\rho_1 - \rho_2}{\mu}\right)$$

Rittingers Formula:

$$v = c\sqrt{d\,\frac{\rho_1 - \rho_2}{\rho_2}}$$

where:

v = Slip velocity, fps

d = Diameter bubble, ft

ρ_1 = Air density, lb per cu ft

ρ_2 = Oil density, lb per cu ft

μ = Oil viscosity, lb per sec-ft units—centipoise $\times$ 0.000672

c = Constant varying from 9 for spheres to 3.5 for flattened bodies

Note that the smaller the bubble, the slower the rise. Because the engine scavenge pump design inherently produces small air bubbles and the air-oil mixture returns to the tank through a tubular air-oil cooler with approximately 0.040-in. oil-passage-tube spacing, it can be concluded that the air bubbles are very small on arrival at the oil tank. A deaerator is therefore needed.

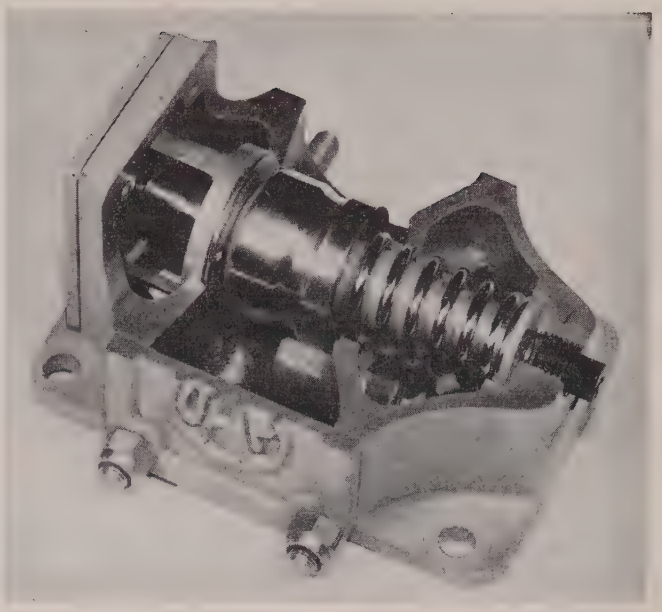

Fig. 3 – U-4990 tank-top-mounted sequence or diverter valve

Over the past 15 years, numerous mechanical deaerators have been devised to remove the entrained air. The deaerators generally required a working pressure and, as such, were located in the engine scavenge oil line between the oil cooler and tank as shown in Fig. 1F,[4, 5] One current USAF cargo aircraft uses a 4.1-lb cast "oil-swirler" type deaerator mounted in the tank top.

With these mechanical devices, it was usually found that the air could be removed down to about 2 to 4%, but when the deaerated oil traveled through the hopper, it picked up air again with a net deaerating gain of little. Antifoaming additives have been used in some cases with success, but apparently the military services have not approved the additives for general usage.

It will be recalled that before the hopper was inserted between the tank in and out ports, a deaerating settling chamber was provided by the tank. With normal temperatures in the tank (170-185 F), the conditions of (a) low velocity, (b) longer oil dwell time, and (c) high oil temperature resulted in effective deaeration of the returned aerated tank oil. The aeration problem then can be solved by devising a method to retain the settling

[3] See Gulf Research & Development Co. Research Project KG 00 (24), September, 1943, "Oil Aeration," by R. J. S. Pigott.

[4] See Air Force Technical Report No. 5764, Feb. 15, 1949, "Aircraft-Engine Lubrication System Deaerator Investigation," by D. J. Gildea and P. H. Schweitzer. (Reprint of a report written by the Engineering Experimental Station of Pennysylvania State College.)

[5] See Boeing Aircraft Laboratory Test of B-17 Oil System Deaerators Model B-17F, Aircraft No. T-21925, Sept. 6, 1944.

Table 1 – Dearation Comparison of Conventional Hopper System versus Diverter System Using Grade 1120 Oil

Oil Tank Tested	Capacity, US gal	Engine Oil Flow, lb/min	Oil Temp, F	Oil Level in Tank	Air, Tank Inlet, %	Air, Tank Outlet, %	Air Removed, %	Remarks
RCAF Lancaster	44	113	180	¾ full	16	12	25	Conventional hopper system
					21	1	95.3	Diverter system
RCAF Lancaster	47	113	180	¾ full	22	12	45.4	Conventional hopper system
					22	0	100	Diverter system
Grumman F8F-1	17	140	167	¾ full	16	7	56.2	Conventional hopper system
					18	3	83.3	Diverter system
RCAF Mosquito	19	100	175	¾ full	23	10	56.5	Conventional hopper system
					24	2	91.7	Diverter system

chamber for normal operating conditions and using the hopper to contain the diluted oil for cold-temperature conditions.

An obvious step is to divert the oil outside the hopper, being careful that the "to tank" line oil impinges tangentially on the tank side. This method returns the deaeration settling chamber required.

This type system was pioneered by United Aircraft Products, Inc., and was specifically mentioned in 1945[6] as a system meriting the serious consideration of the aircraft industry. The diverter system was used by the Navy as early as 1943 by mounting on the tank top a thermostatically operated oil-tank diverter valve (or sequence valve), shown in Fig. 3, sensitive to the returned oil temperature. Below 130 F the oil was directed through the hopper to aid in fast oil temperature warmup. Between 130 to 160 F the sequence valve directed the return flow partly to the hopper and partly outside the hopper (called "to tank line"). Above 160 F, all returned oil was diverted outside the hopper – "to tank." As usually occurs in the early phases of an engineering program, problems arose. The diverter-valve problem was that the valve sensing tank return oil temperature diverted oil after a short duration of engine running time which resulted in colder make-up oil being forced into the engine. The colder oil returned to the valve resulted in directing the oil flow back to the hopper. This cycling action continued until the entire oil supply was warmed.

After reaching a constant normal tank oil temperature of 170-185 F, however, the aerated oil was effectively deaerated. A later diverter valve was then designed which is mounted on the tank bottom (Fig. 4). This valve has been, and is, currently installed on several different type production Navy aircraft. Both the engine supply line and scavenge line are connected to the diverter valve at the tank bottom. The "to hopper" and "to tank" lines are routed inside the tank, both opening inside the tank at the top (Fig. 5). The thermostatic element now senses a mixture of make-up and hopper oil. This arrangement eliminates the cycling condition formerly encountered. A special adaptation of this valve is now used on the new Lockheed Navy P2V-6 aircraft.

A motor-operated diverter valve mounted in the tank top, sensitive to a temperature pickup in the engine supply line, was used experimentally. However, the complications and weight of a motor with wires, lights, switches, and the like resulted in the abandonment of this type valve for production aircraft.

In developing the principle of using a diverter valve, an extensive program was conducted at United Aircraft Products, Inc. Table 1 is a tabulation of some of the deaeration results obtained on aircraft oil tanks on which a diverter valve was mounted. Note the gain in deaeration obtained by using a diverter system. Fig. 6 shows a simulated oil system used by the Powerplant Laboratory, Wright Air Development Center, to demonstrate the effect of a diverter system. The engineer is pointing to a sight glass in the tank oil out line. Fig. 7 shows the sight glass before, and Fig. 8 shows after the oil was diverted. Again note the reduction in aeration obtained simply by diverting the oil outside the hopper. Further, a flight test conducted in 1951 on a military bomber using the diverter system showed an increase of 15 psi main engine oil pressure over the conventional hopper system at 40,000 ft altitude. Special note should

[6] See *SAE Journal*, Vol. 54, No. 4, April, 1946, pp. 174-190: "Factors in Aeration and Deaeration of Aircraft-Engine Oil," by W. L. Weeks.

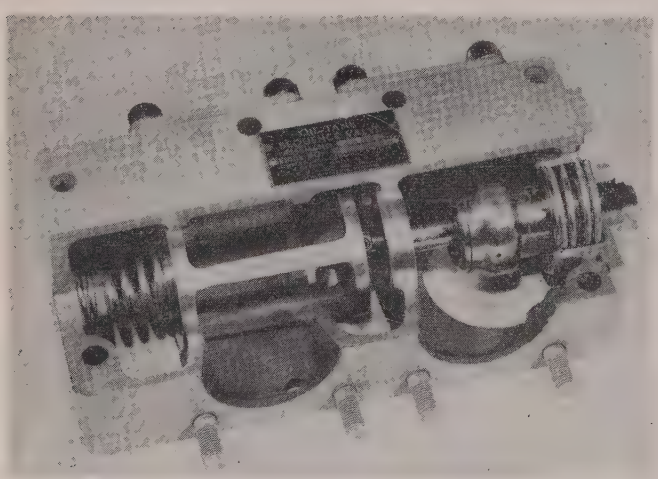

Fig. 4 – U-6990 tank-bottom-mounted sequence or diverter valve

be given to the fact that with a diverter system, *all* the oil is circulated through the engine instead of the small volume contained in the hopper system only. Further advantages of the diverter or de-aerating system as applied to cold-weather operation will be discussed later.

Cold-Weather Oil System

In the case of Navy aircraft, there is no –65 F engine-start requirement, hence the cold-weather oil-system problem has not been a Navy consideration.

In the case of the USAF and commercial aircraft, the basic cold-weather oil-system problem has always been that the undiluted make-up oil was not positively segregated from the diluted hopper oil. After engine shutdown and exposure of the aircraft to cold temperatures, the heavier undiluted oil seeped into the hopper, settled in the tank out line and even further downstream in the engine supply line. The undiluted oil froze, thus plugging the engine oil supply line. The result could be, and was many times, the loss of the engine after start. To preclude such a possibility, the USAF Standard Operating Procedure is to apply externally supplied ground equipment heat for from 1 to 3 hr to the engine and oil system. Conservatively, in every case, 80% of this time is necessary for the oil system only. Considerable development work[7] has been done to obtain a system where *positive segregation* of the two oil bodies in the tank could be obtained. This would eliminate the need for heat and, more important, the *time* required to preheat the oil system before an engine start.

With the hopper system, rapid warmup of the engine-aircraft circulating oil system is obtained. But the additional problem of adequate warmup of the frozen make-up oil during flight has generally not been sufficiently emphasized. It is known that the delayed warmup of the make-up oil can result in engine-oil starvation with resultant loss of the aircraft and possibly the crew when operating in the arctic regions.

As stated previously, considerable work has been accomplished to obtain an oil system design to provide positive segregation of the undiluted make-up oil from the diluted hopper oil. The first approach made was to bolt a standard motor-operated shutoff valve to the hopper in place of the hopper flapper valves. However, installation and operational problems discouraged this idea. Then tests were conducted on a design where a hopper poppet-type valve was connected to a shaft, which in turn was connected to a motor actuator. Freezing of the shaft and linkage discouraged this method of attack. To eliminate the possibility of the motor failing to operate after a cold-temperature soak,

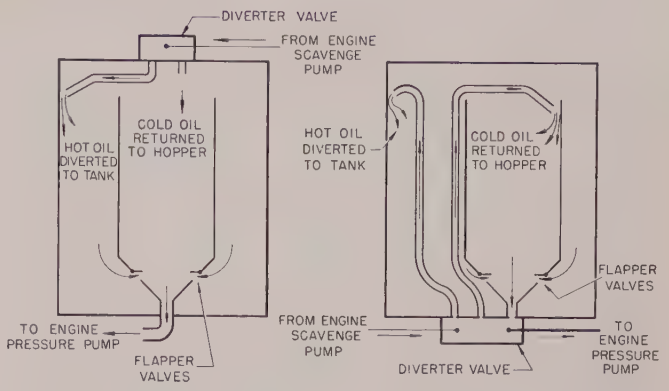

Fig. 5 – Schematic of tank-top- and tank-bottom-mounted sequence or diverter valve

[7] See references 8-13 under "Additional References" at end of this paper and also reference given in footnote 9 of this paper.

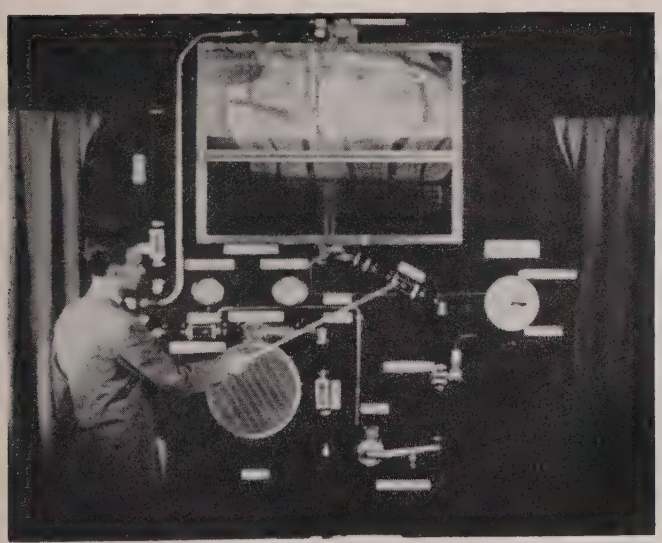

Fig. 6 – WADC powerplant laboratory simulated engine oil system

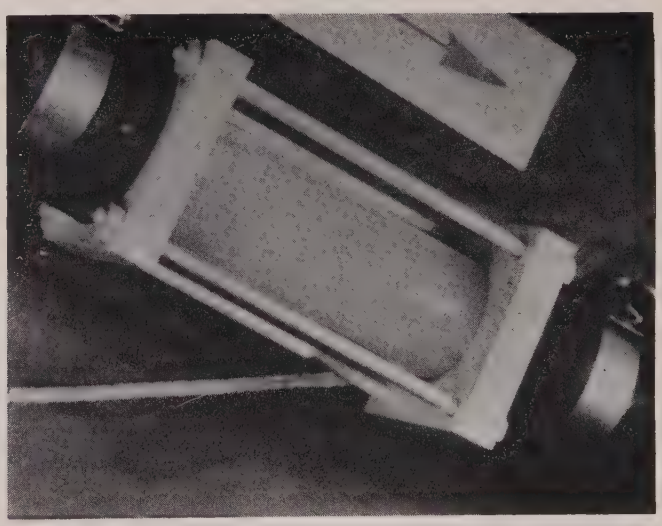

Fig. 7 – Tank oil out line sight glass before oil is diverted

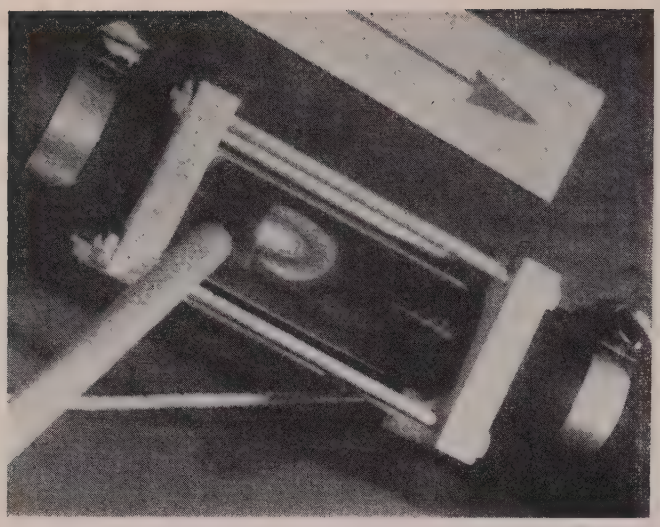

Fig. 8 – Tank oil out line sight glass after oil is diverted

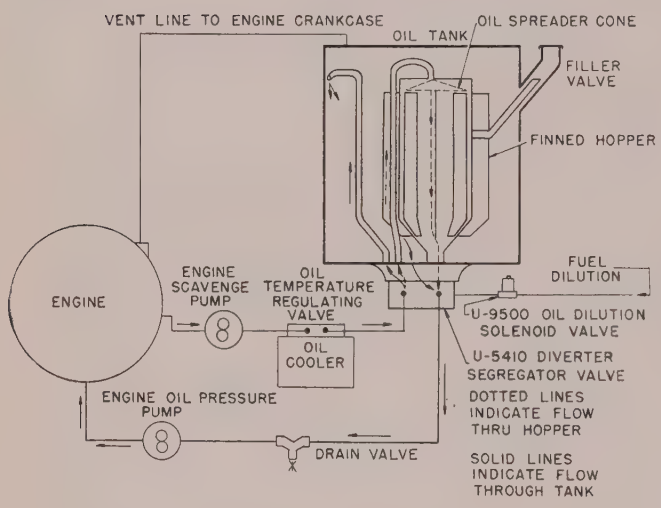

Fig. 9 – Diverter-segregator finned-hopper oil system

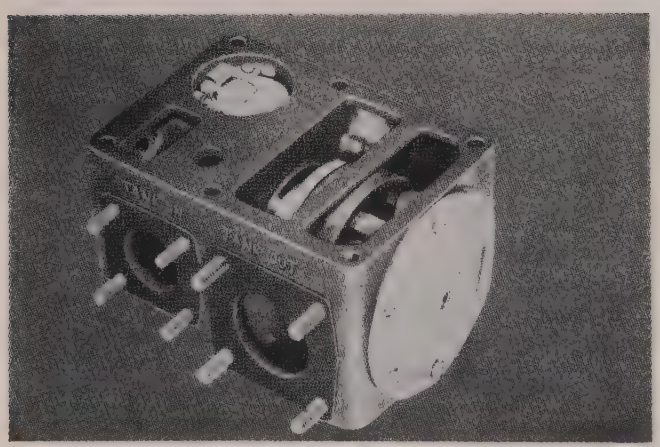

Fig. 10 – Diverter-segregator valve

a valve was designed to be "cocked" by an actuator in the closed position at engine shut-down. After engine start, it was planned for the gradual decongealing of frozen oil to permit the segregator valve to open because of a cocked or preloaded torsional spring. The development of this valve was abandoned by the designer.

In every case, due to the possibility of failure of the electrical system, it was necessary that the actual valve position be indicated by a series of lights with instructions to the pilot not to take off until certain lights went off and others came on.

Because of the complexity of the electrically operated valve approach, USAF engineers at the Wright Air Development Center were very interested when a modified thermostatically operated United Aircraft Products diverter valve was proposed to provide the desired positive segregation. In addition, UAP demonstrated to the USAF a specially designed finned hopper to provide fast warmup of the frozen make-up oil surrounding the hopper.

The solution to the cold-weather oil-system problem has since been determined to be the use of an automatic, mechanically operated diverter-segregator valve in combination with a specially designed "heat-transfer" hopper.

Deaerating Cold-Weather Oil System

Simultaneously, in late 1949, a valve design incorporating positive segregation was presented to the USAF, and an oil decongealing demonstration of a "heat-transfer" finned hopper at –65 F was made. The combination of a diverter-segregator valve and finned hopper received the immediate and enthusiastic support of the USAF. In fact, the application of the system to a B-50 was supported by the USAF.

During the winter of 1950-51, the system in principle was tested in Alaska and satisfactory results obtained on a Grumman Arctic Rescue SA-16 aircraft. These results gave further impetus to the project so that by early 1952, the diverter-segregator finned-hopper oil-system development was completed. This system consisted of a special finned hopper inside the tank and the diverter-segregator valve mounted on a combination sump adapter outside the oil tank, as shown in Figs. 9 and 10. Except for vent ports at the top of the hopper, there is no interconnection of hopper and tank. Valves within the diverter-segregator assembly, actuated by hydraulic or spring forces and a thermostatic control, regulate the oil flow into and out of the hopper and tank (Fig. 11). The hopper is supported in the sump adapter by the "to hopper" and "to tank" tubes (sealed by "O" rings). "Hopper out" and "tank out" ports of predetermined areas are also located in the sump adapter.

Diverter-Segregator Valve and Finned-Hopper Combination

Figs. 10 and 12 show that the U-5410 diverter-segregator valve in principle is similar to the

U-6990 diverter valve of Fig. 4, with a means for providing positive segregation added. The "to hopper" and "to tank" lines are located similarly to the U-6990 diverter valve installation previously described. The operation of this diverter-segregator valve is mechanical and completely automatic. Positive segregation between the two bodies of diluted and undiluted oil is always provided by rubber seals which prevent seepage of undiluted oil into the tank out line; hence no frozen plug will exist, and a supply of diluted oil is provided for cold-temperature engine starts without the use of preheat, which assures that engine crankability will be provided.

Immediately on engine start, duplicating the conventional oil system make-up oil supply provisions, the make-up oil supply port is preopened to pass make-up oil to the circulatory system as it melts and is demanded by the engine. A minimum hydraulic force of 2 psi is necessary to provide this preopening. Therefore, to assure, regardless of engine speed, that the 2-psi minimum pressure will be provided at the diverter-segregator assembly "in" port, a preloaded 2-psi flapper valve is located in the "to hopper" line immediately above the diverter-segregator valve. This flapper valve also prevents back flow to the engine of the oil contained in the "to hopper" tank system. During the oil temperature warmup to 130 F, the cold oil is directed only to the hopper. The "to tank" port remains closed. See Fig. 11.

Heat rejected by the engine raises the oil temperature, and when the thermostat reaches 130 F, the diverter valve starts opening the "to tank" port permitting oil to start to flow into the tank but only after the oil in the tank line has been sufficiently warmed to allow flow. Should the "to tank" line still be congealed when the diverter valve has moved to the "to tank" position, the valve will relieve at 12 psi and continue to direct flow to the hopper only until the "to tank" line clears. Simultaneously, heat rejected to the oil flowing through the engine is transferred through the special hopper melting the surrounding undiluted oil in the tank.

When the thermostat reaches 160 F, the diverter valve will have moved to the full open tank port position, while nearly closing the "to hopper" port (Fig. 13). A small quantity of oil always flows through the hopper to prevent entrance of air at the hopper outlet. With the thermostat located in the out chamber, where hopper and make-up oil is mixed, engine oil inlet temperature fluctuation is avoided. Temperature is steady during the slow rise up to normal operating temperature.

If inlet-oil temperature rises above the normal operating level – due to engine operation – the additional expansion of the Vernatherm thermostat is absorbed by compression of an integral relief spring, without changing the position of the diverter valve.

Prior to engine shutdown in cold weather, fuel is directed into the diverter-segregator valve under

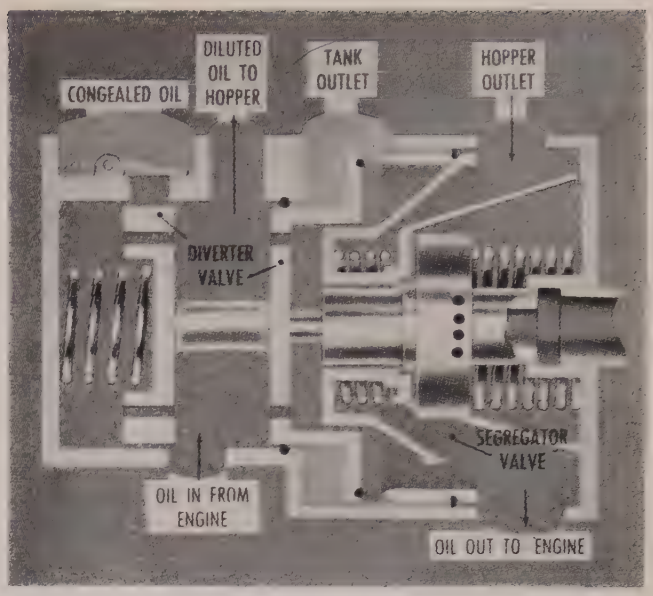

Fig. 11 – Diverter-segregator valve schematic in engine oil system starting "warmup"

pressure. The fuel (even at 70 F temperature) flowing through and around the thermostat results in the thermostat's retracting in 2 to 5 sec (Fig. 14). The return spring forces the diverter valve to close the "to tank" port and direct the oil flow through the hopper, resulting in the dilution of the oil in the hopper circuit. The preopening of the segregator valve allows a little oil at the tank bottom to be slightly diluted, which diluent rises to the top of the tank after the segregator valve

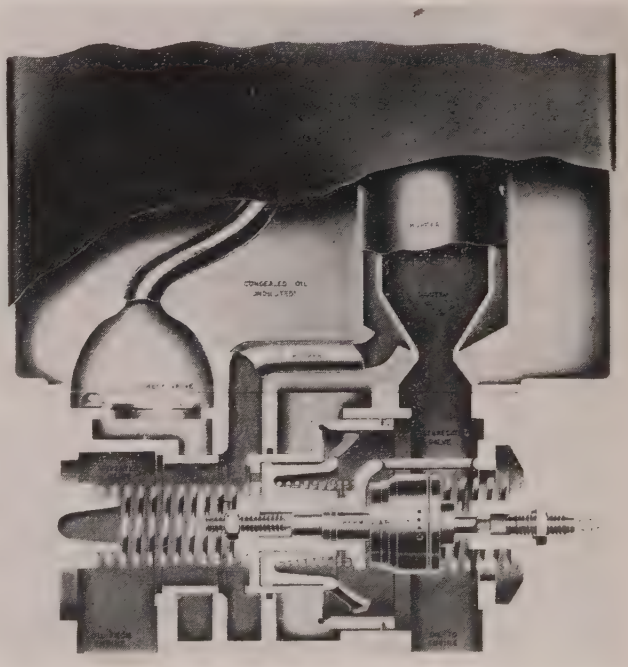

Fig. 12 – Deaerating cold-weather oil system after engine shutdown

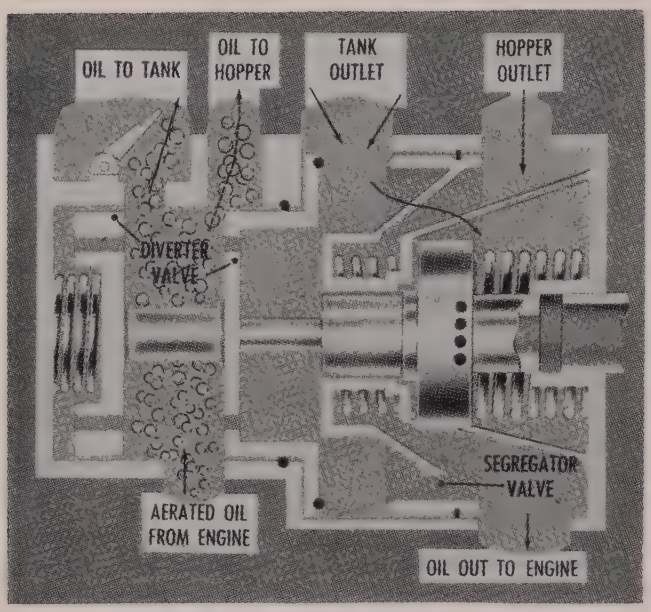

Fig. 13 – Diverter-segregator valve schematic in engine oil system "warmup complete" position

closes at engine shutdown. Complete segregation – closed check and segregator valves – prevents the collection of undiluted oil in any portion of the hopper circulatory system.

As the thermostat absorbs heat from the surrounding warm oil, after oil flow has ceased because of engine shutdown, the diverter valve tends to move away from the segregator valve. Opening of the segregator does not occur, however, as it is forced to follow the diverter by the force of the segregator spring. The diverter-segregator valve assembly weighs 7.2 lb.

Since there is no communication between the hopper and tank oil supplies, it is necessary to provide a means to fill selectively either the hopper or oil tank from a single point. This is done simply by inserting a hopper filler tube inside the normal tank filler. A snap-lock-type cap over the hopper filler tube is a simple and easy method of sealing the hopper filler tube.

Discussion of System Tests and Test Results

The UAP and USAF flight tests were conducted with MIL-0-6082 grade 1100 oil. UAP tests consisted of functional tests on the diverter-segregator valve and performance tests of the valve and hopper combination at various tank levels.

Several design changes were made to the diverter-segregator valve in order to reduce pressure losses. Figs. 15 and 16 show oil-pressure data. Since the valve is a new item in the engine oil supply line, it is in order to discuss the pressure losses.

First, it must be remembered that the system *actually* deaerates the engine supply oil down to 2 to 4% (by volume). Conservatively, tank return oil aeration is 15% at low altitudes and higher with increasing altitudes.[4, 8] Referring to the Pigott report,[3] where it is stated that the aeration effect on the engine oil pump effectively reduces the pump volumetric efficiency, one can state that by preventing this efficiency loss, with a slightly lower pump absolute inlet pressure due to the pressure drop through the valve, the *actual* weight of oil pumped through the engine would be essentially the same with the deaerating cold-weather system as with the conventional system.

Further note that the aircraft-engine pressure

[8] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 519-529: "High-Altitude Aircraft Oil Systems," by F. E. Carroll, Jr.

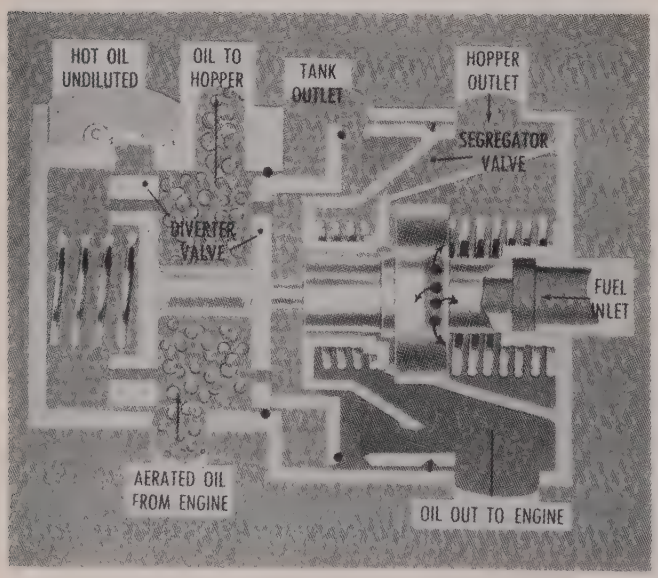

Fig. 14 – Diverter-segregator valve schematic in engine oil system "during dilution" position

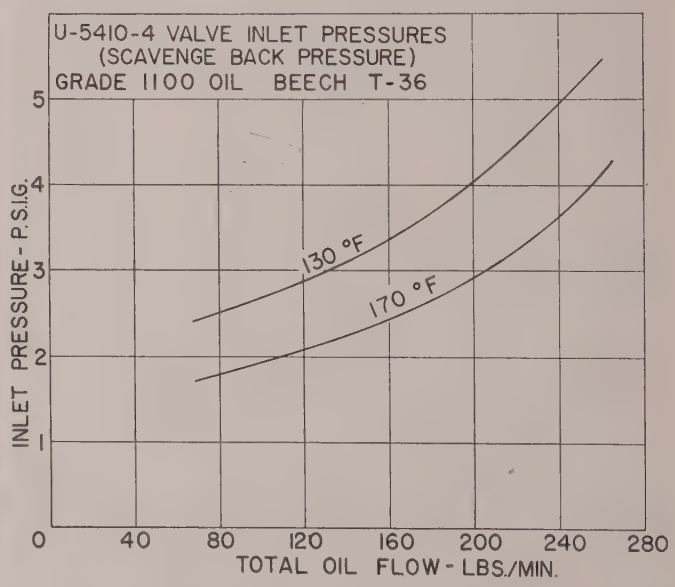

Fig. 15 – U-5410-4 valve inlet pressures (scavenge back pressure)

pumps are provided with from 40 to 50% greater capacity than the maximum demand of the engine. The excess is normally bypassed around the pump so that the full capacity of the pressure pump is never delivered to the engine except at extremely high altitudes. Also, the pressure pump is designed to provide the required main oil pressure at an absolute inlet pressure of 8 in. of Hg and the minimum required oil pressure at an absolute inlet pressure of 6 in. of Hg. Therefore, since the pump oil supply is now deaerated, the deaerating cold-weather oil system oil-pressure losses are of little significance when comparing the overall system performance with the conventional system performance. It will be noted from Fig. 15 that the system "in" pressure (engine-scavenge-pump back pressure) is well below the maximum of 40 psi to which the scavenge pumps are guaranteed, allowing of course for an additional back pressure of 15 psi for the oil cooler and 4 psi for line losses.

Figs. 17 and 18 show a typical group of curves covering results from typical cold-temperature tests. Several endurance tests were conducted at normal operating temperature with a supply of dirty oil furnished by the Wright Air Development Center. No detrimental effects were evident with special attention given to the diverter-segregator valve. Numerous tests have been conducted respectively at a make-up undiluted oil temperature of −10, −20, −30, −40, and −60 F or lower temperatures. These tests in all cases consisted of a setup essentially as shown in Fig. 19. The procedure was to run at normal idling oil flow, dilute through the standard diverter-segregator-valve fuel connection, and shut down. Room-temperature fuel was introduced at a rate and pressure simulating the actual aircraft fuel dilution system performance.

The entire hopper, make-up oil, diverter-segregator valve, and approximately 2 ft of the engine supply line were maintained in a cold dry-ice fuel

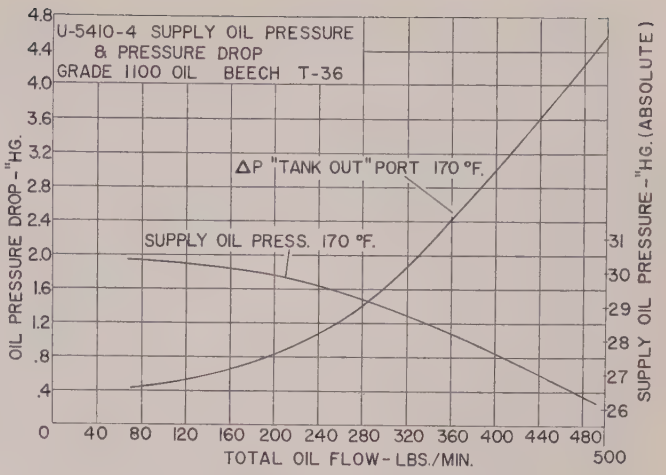
U-5410-4 SUPPLY OIL PRESSURE
& PRESSURE DROP
GRADE 1100 OIL BEECH T-36
ΔP "TANK OUT" PORT 170 °F.
SUPPLY OIL PRESS. 170 °F.
OIL PRESSURE DROP - "HG.
SUPPLY OIL PRESSURE - "HG. (ABSOLUTE)
TOTAL OIL FLOW - LBS./MIN.

Fig. 16 – U-5410-4 supply oil pressure and pressure drop

bath between −65 to −75 F until the undiluted frozen make-up oil temperatures had reached −60 F or the required test temperature. The starting procedure established in joint meetings with the Wright Air Development Center engineers was to run for 20 min at engine-warmup-speed oil flows (1000-1200 rpm) and then to increase oil flow to engine take-off speed. An oil-temperature warmup versus time curve was followed as furnished by the WADC engineers. The engine-oil-temperature warmup curve, Fig. 20, is based on actual data obtained in service or tests by the USAF. Note that at 20 min the oil temperature is only 104 F, yet in every case the frozen make-up oil has been melted at this point. By following the curve, full oil flow was diverted at 24½ to 28 min, depending on the particular system being tested.

In all cases positive segregation was provided. Preheat was never used. Further, depending on the system being tested, decongealed make-up oil surrounding the fins could be seen at 8 to 12 min from

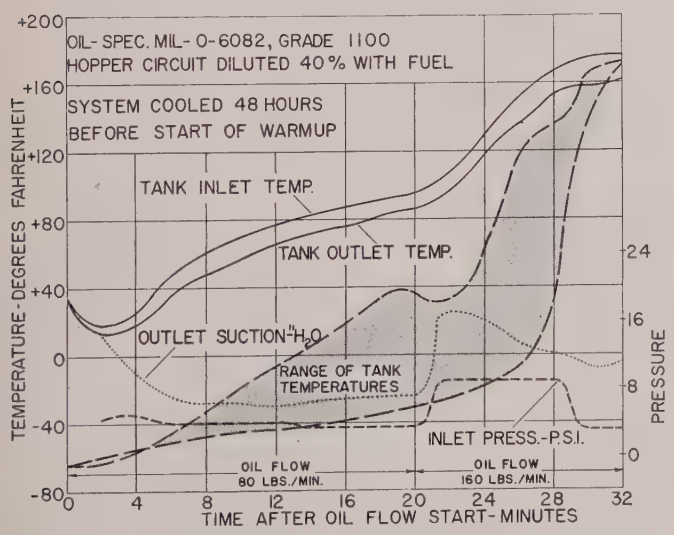

Fig. 17 – Cold-temperature engine-start test data

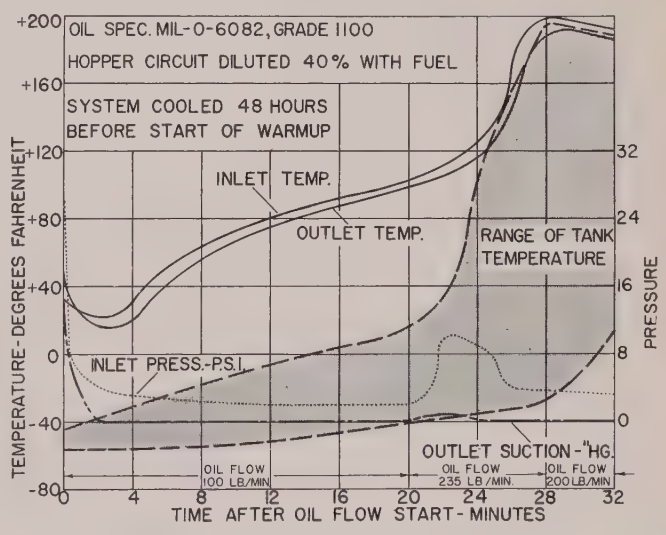

Fig. 18 – Cold-temperature engine-start test data

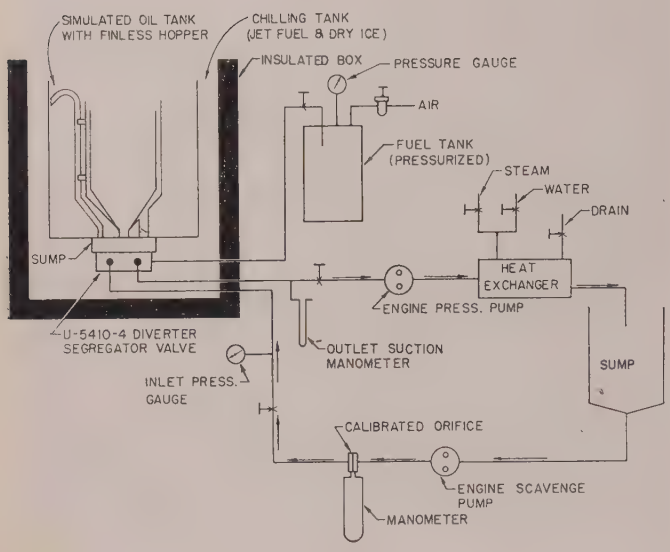

Fig. 19 – Schematic diagram of laboratory setup for decongealing tests

start. When the oil flow was increased at 20 min to the take-off rating, the decongealed make-up oil immediately entered the circulating system through the segregator preopening. This could be noted by observing the empty annulus in the frozen oil and by noting the increase in hopper level. The 5/16-in.-wide annulus around fins and hopper was more than required to pass the diverted oil flow. On continuing to run, the frozen make-up oil is melted by the diverted oil flowing through the annulus. Fig. 21 shows a typical result after running for 30 min. Special attention should be given to this feature where the frozen make-up oil continues to be melted by the diverted oil flow.

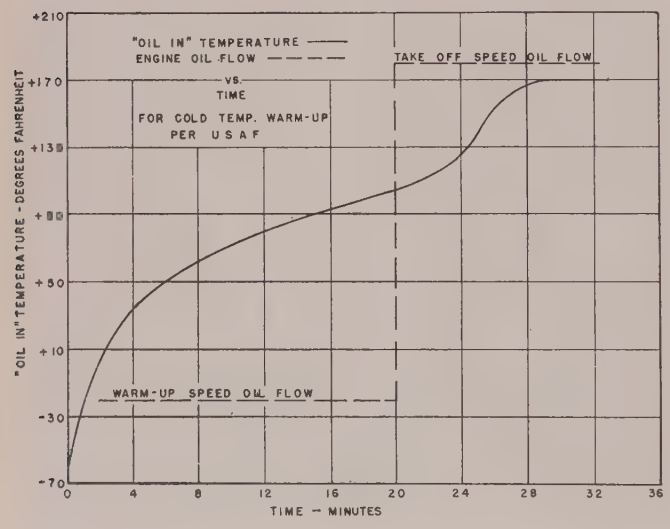

Fig. 20 – Engine oil flow and oil-temperature warmup versus time curve used for all cold-start tests

A detailed analysis of the use of the diverter-segregator valve design by aircraft and USAF design engineers has disclosed that the design inherently includes "fail-safe" provisions. The hopper outlet is always open; the diverting action directing oil in the tank (outside the hopper) and the opening of the "from tank" port are coupled integrally in a "spool" arrangement; hence, if the oil is diverted, the tank out port is open. Further, independent of the oil temperature immediately on engine start, the segregator preopening enables make-up oil to enter the circulatory system.

The Vernatherm-type thermostat used in the diverter-segregator valve is the same type used in air-oil coolers produced since approximately 1944 and in all jet-engine fuel-oil coolers. This thermostat is also used in numerous commercial applications. In all cases, the Vernatherm has proven to be highly reliable. However, for purposes of analysis, it should be noted that if thermostat failure occurs, the engine oil supply continues to be available. Likewise, if springs fail, the engine oil supply continues to be available. A failure or combination of failures will result only in not providing positive segregation. In principle, the diverter-segregator valve incorporates a production-quantity service-tested diverter valve "spool" design.

Tests have been conducted on B-36, B-50, SA-16, T-36, H-21, and C-123 systems. The system is now in production on the Grumman SA-16 and Piasecki H-21. Production is scheduled for the Beech T-36 and Chase C-123. Further, the B-50 system was installed on one engine of a Fairchild C-119 and successfully service tested in Alaska this past 1952-1953 winter. An additional Beech T-36 system, although designed for the Pratt & Whitney R-2800-99 engine, was successfully tested on a Wright R-2600 engine ground rig by the RCAF in Churchill, Canada, the latter part of this past winter. Temperatures in Alaska were as low as −50 F and down to −30 F in Churchill. As a matter of interest, the system is being installed experimentally on two Army tanks using dry-sump engines.

Further Hopper Development

After an aircraft has landed and a cold temperature is anticipated while running at engine idling speeds, the diluting procedure is to inject fuel into the oil system for a given period of time to obtain the required per cent oil dilution. It is desired that the time required to dilute the oil be held to a minimum due to engine considerations. In a conventional system the diluted mixture overflows from the hopper which in turn forces undiluted make-up oil into the circulating system with this action continuing. Hence, generally to obtain 40% dilution requires time in excess of 10 min. It is known that in some systems it actually is impossible to obtain the 40% dilution since after extended periods of engine running (15-20 min), the fuel boils off because of engine heat rejected to the fuel-oil mixture. A USAF hopper design was therefore

proposed with an additional volume provided at the hopper top in the shape of a mushroom to contain the added fluid volume. This design was used on the first UAP hoppers. But since experience has shown that during the introduction of fuel into the diverter-segregator valve a very small amount of fluid "backflow" through the make-up oil port takes place which prevents undiluted make-up oil from entering the system (during injection of fuel only), the need for containing the added fluid in the circulatory system is no longer necessary – when the hopper overflows, make-up oil cannot enter the circulatory system with the deaerating cold-weather oil system.

Keeping in mind the objective of holding the time to a minimum for diluting the oil system, the following analysis shows that it will take less time to obtain a given per cent dilution without a hopper expansion tank than with a hopper expansion tank. In order to visualize the system where an expansion tank is not used, a sketch is provided in Fig. 22. The term "virtual volume" refers to the total fuel and oil volume which would exist if there was no hopper overflow. The term circulatory system refers to the total volume of oil contained in the hopper, oil lines, engine, oil cooler, and so forth.

Fig. 21 – Decongealed annulus provided 30 min from start of —60 F test on Beech T-36 diverter-segregator finned-hopper oil system

Without Hopper Expansion Tank

v = Volume of oil in circulatory system, gal
q = Dilution rate, gpm
t = Dilution time, min
Δt = Time increment
V_{tv} = Total virtual volume, gal.
Q_{tv} = Diluent volume obtained in Δt, gal
% = Dilution per cent

From equation (1)

$$\% = 100 \times \frac{Q_{tv}}{V_{tv}}$$

$$Q_{tv} = q\Delta t$$

$$V_{tv} = v + Q_{tv} = v + q\Delta t$$

With Hopper Expansion Tank

v = Initial volume of circulatory system oil, gal
q = Dilution rate, gpm
t = Dilution time, min
Q = Diluent volume, gal
V_t = Total volume (mixture) at time t, gal
% = Dilution per cent

$$\% = 100 \times \frac{Q}{V_t} \qquad (1)$$

$$Q = qt \qquad (2)$$

$$V_t = v + Q$$
$$= v + qt \qquad (3)$$

$$\% = 100 \times \frac{qt}{v + qt}$$
$$= \frac{100}{\frac{v}{qt} + 1} \qquad (4)$$

Using an increment of one min, a dilution curve with hopper overflow can be determined by continuing calculations as follows:

At end of first minute:		At end of first minute:	
Initial volume of circulatory system undiluted oil	v = 10 gal		
Added diluent	= 1 gal	Added diluent	= 1 gal
Total volume	= 11 gal		

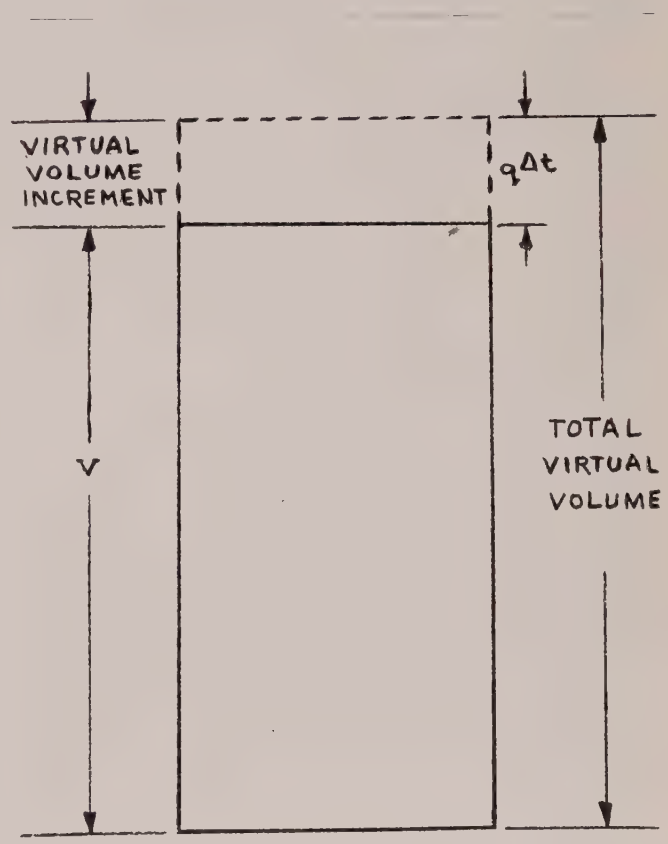

Fig. 22 – System without hopper expansion tank

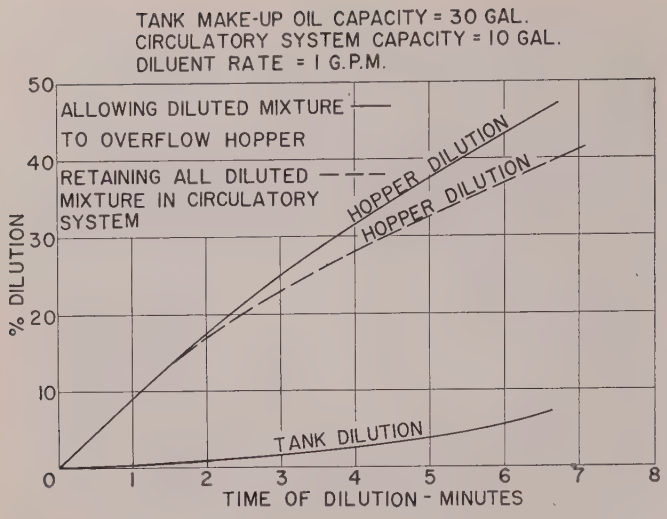

Fig. 23 – Comparison of circulatory oil system oil dilution versus time for a hopper with and without an expansion tank

Without Hopper Expansion Tank		With Hopper Expansion Tank	
From equation (1):		From equation (4):	
Dilution	= 9.09%	Dilution	= 9.09%
Diluent in 10-gal mixture	= 0.909 gal		
At end of second minute:		At end of second minute:	
Initial volume of 9.09% diluted oil	v = 10 gal		
Added diluent	= 1 gal		
Total virtual volume	V_{tv} = 11 gal		
Diluent in 11 gal	= 1.909 gal		
From equation (1):		From equation (4):	
Dilution	= 17.35%	Dilution	= 16.7%
Diluent in 10-gal mixture	= 1.735 gal	Diluent in 12-gal mixture	= 2 gal

The calculations can be continued. Fig. 23 plots this data up to 40% dilution. In addition, for an oil-tank make-up-oil volume of 30 gal, the actual per cent dilution of the make-up oil is shown. Note at 40% dilution of the circulatory system with no expansion tank, the make-up-oil dilution is only 6%. Further, it should be recalled that in the conventional oil system the make-up-oil dilution was always considerably higher (20%)[9]. The expansion-tank weight on an R-4360 engine system (Fig. 24A) is 4.2 lb and on an R-1820 engine system is 1.4 lb (Fig. 24B).

Since the time to obtain a given per cent dilution is less without the expansion tank and the expansion tank adds weight to the aircraft, it is concluded that the expansion tank should not be used.

In general, a finned hopper weighs more than the standard oil system hopper. Further, recent experience has shown the installation problems presented by aircraft manufacturers and the USAF for the system are for retrofits on existent tactical aircraft. Retrofitting involves discarding the sump and hopper now installed and replacing with a sump adaptor, a finned hopper, and adding a hopper filler tube.

However, in some cases it is impossible to insert a finned hopper in existing tanks. Therefore, to obtain a lighter hopper and an adaptable design for retrofit on aircraft already produced, a further development of the hopper was initiated. Now completed, the result is the double-wall or finless hopper. A comparison of the Chase C-123 finned hopper with the finless hopper is shown in Fig. 25. As in all cases, an appreciable weight saving is effected.

In using the finned hopper, a considerable amount of surface was used effectively to transfer heat. The heat from the oil transferred through the fins rapidly melts the frozen make-up oil.

The finless hopper design consists of one hopper being placed inside a larger-diameter hopper. A space is left between the two and held constant by spacers. The "to hopper" tube is connected to the annulus between the hoppers and the "to tank" line is routed on the side of the outer hopper as before (Fig. 26). To obtain effective heat transfer with the finned hopper, considerable fin area was used. To obtain the heat transfer and, as it will be proven, improved heat transfer from the finless hopper, the velocity of the effective heated oil is increased past the metal adjacent to the frozen oil, thereby considerably increasing the oil-film coefficient, and hence the heat-transfer coefficient. Thus, in the general equation,

$$H = UA\Delta T$$

where:

H = Heat transfer, Btu per min
U = Heat-transfer coefficient
A = Area, sq in.
ΔT = Oil-temperature difference

In the case of the finned hopper, the value of A is very high while the heat-transfer coefficient is low. In the case of the finless hopper, the area A is no greater than a standard hopper but the heat-transfer coefficient is greatly increased, and hence, the heat-transfer rate is increased.

It has been demonstrated by actual tests on a finned hopper and a finless hopper for the same installation running the test exactly the same, that the heat-transfer characteristics of the finless hopper design are better than those of the finned hopper.

An analysis can be made showing the reason for the improved performance. That is, the time required to decongeal the weight of oil contained in a ½-in.-wide annulus around the fins and hopper

[9] See SAE *Quarterly* Transactions, Vol. 6, April, 1952, pp. 175-195: "Low-Temperature Lubrication of Aircraft Engines," by Saul Barron.

for a standard finned hopper and the time required for a finless hopper to decongeal the same weight of oil shall be calculated.

For the finned hopper a standard design of 10 fins extending 2 in. inside and 2 in. outside (0.040-in. aluminum) will be used. For this analysis, both the finless and finned hoppers shall be of 4½-gal capacity, 10 in. in diameter, and 15 in. high, and the objective shall be to obtain a ½-in.-wide decongealed annular path around the finned hopper.

Since the basic problem here presented is one of unsteady-state heat transfer, it is necessary that certain assumptions be made. It shall be assumed that the oil does not flow until 10 F is reached (oil pour point). The cold-oil heat-transfer coefficient shall be assumed to be equal to the hot-oil coefficient. An oil-film coefficient value of 0.003 Btu sec/ft²/F (for stagnant oil based on UAP oil-cooler design experience) and a conservative constant hopper oil temperature of 70 F shall be used in the calculations.

To obtain the decongealed annulus, it is necessary to heat the −65 F oil to +10 F (pour point) giving an average temperature of −27.5 F. To find the rate of heat transfer for the assumed conditions, the following basic equation is used:

$$H = \frac{T_1 - T_2}{\frac{1}{h_1(A_1' + R_1 A_1'')} + \frac{1}{h_2(A_2' + R_2 A_2'')}} \qquad (5)$$

$$R = \frac{\text{Tan } h\, U}{U} \qquad (6)^{10}$$

$$U = l\sqrt{\frac{2h}{ky}}$$

where:

H = Heat transfer, Btu per sec

T_1 = Hopper oil temperature, F

[10] See p. 329 of *Journal of Research of National Bureau of Standards*, Vol. 16, 1921-1922, in paper No. 211, "Radiators for Aircraft Engines," by S. R. Parsons and D. R. Harper, 3rd.

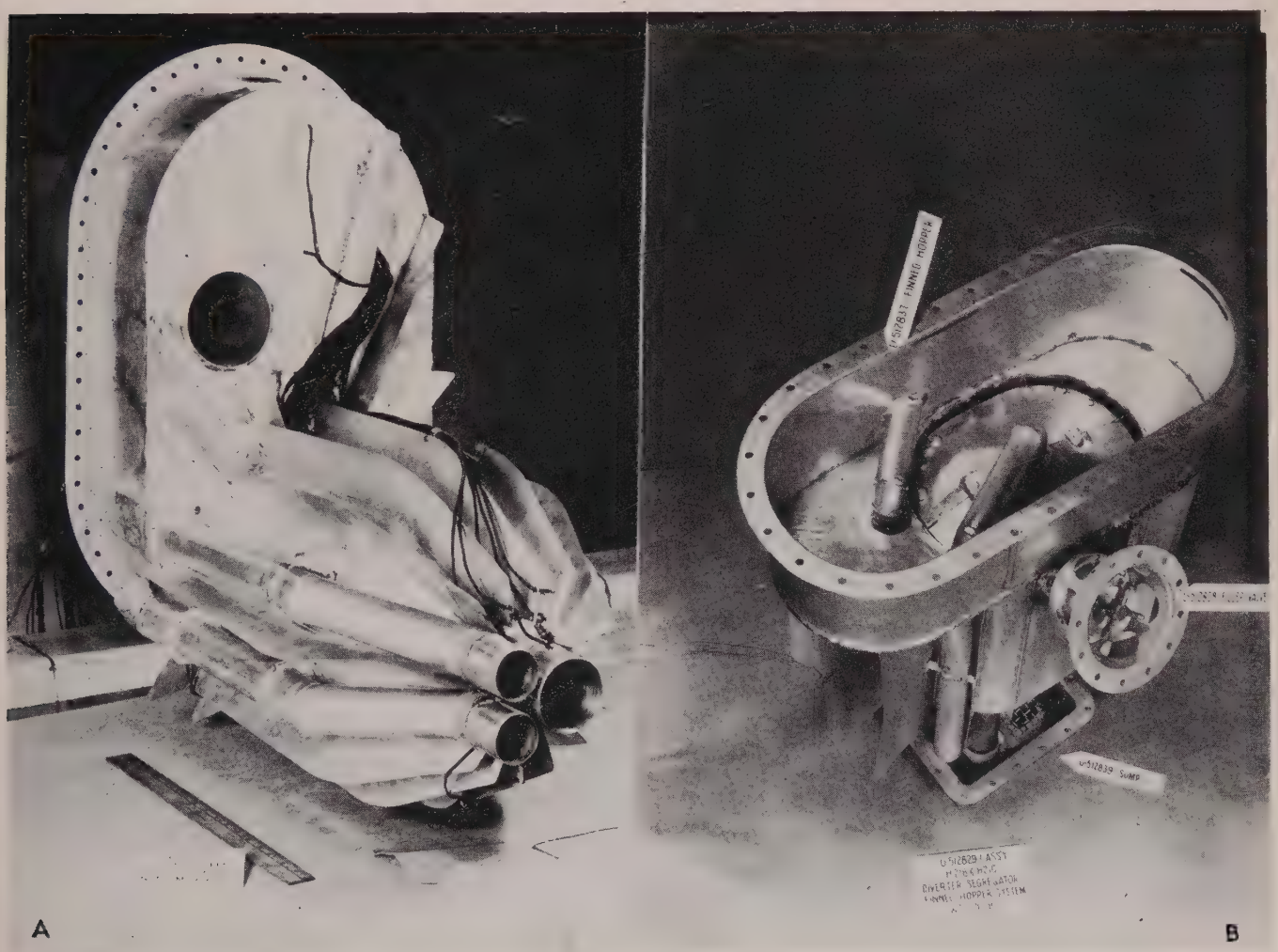

Fig. 24 – (A) B-50, R-4360 finned hopper with expansion tank, and (B) H-21, R-1820 finned hopper with expansion tank

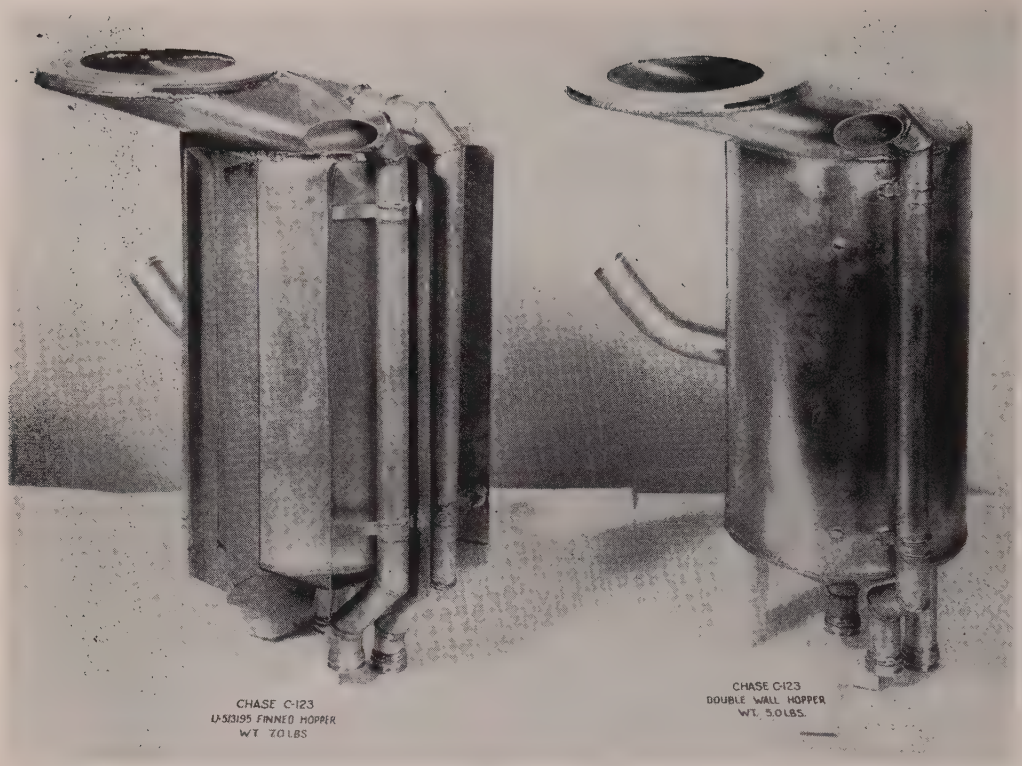

Fig. 25 – Chase C-123 U-513195 finned hopper and Chase C-123 double-wall hopper

T_2 = Average annulus oil temperature, F
R = Fin efficiency, %
h = Oil-film coefficient, Btu/sec/sq ft/F
k = Material conductivity, Btu/sq ft/F/ft/sec
l = Fin length, ft
y = Fin thickness, ft
A' = Primary surface, sq ft
A'' = Secondary surface, sq ft

The calculated heat-transfer surface for the finned hopper is:

$A_1' = A_2' = 3.27$ sq ft

$A_1'' = A_2'' = 4.16$ sq ft

From equation (6) the calculated fin efficiency is:

$R_2 = R_1 = 52\%$

For finless hopper:

$A_1' = A_2' = 3.27$ sq ft

$A_1'' = A_2'' = 0$

$R_2 = R_1 =$ No fins

Since the finless-hopper wall thickness is only 0.040 in., conductivity losses shall be neglected. The volume of oil in a ½-in. annulus for the above described finned hopper is 0.303 cu ft. Assuming a 70 F average oil temperature, this volume oil at a specific gravity of 0.90 weighs 17 lb. Using a specific-heat value of 0.400 Btu/lb/F, the heat required to raise the oil to +10 F is:

$$H_3 = W \times C_p \times \Delta t$$

where $\Delta t = 75$ F

$$H_3 = 510 \text{ Btu}$$

For both of the following cases, $T_1 = 70$ F, $T_2 = -27.5$ F

Finned Hopper Case	Finless Hopper Case
	Providing an annulus between the walls to obtain a mass flow of 56 lb/sq ft/sec and using data obtained from UAP oil-cooler design curves
$h_1 = 0.003$ Btu/sec/sq ft/F	$h_1 = 0.034$ Btu/sec/sq ft/F
$h_2 = 0.003$ Btu/sec/sq ft/F	$h_2 = 0.003$ Btu/sec/sq ft/F (stagnant oil)
From equation (5):	From equation (5):
$H_0 = 0.795$ Btu per sec	$H_1 = 0.878$ Btu per sec
The time required to melt 17 lb of oil with the finned hopper is:	The time required to melt 17 lb of oil with the finless hopper is:
10.7 min	9.7 min

These calculations were carried further, and the data are plotted in Fig. 27 showing the oil weight versus time decongealing characteristics of the finned hopper and the finless hopper.

As previously mentioned, test results have actually shown the finless hopper provides superior decongealing performance over the finned hopper. Fig. 28 shows the decongealed annulus provided in frozen −55 F oil on the Beech T-36 finless hopper at the end of 21 min immediately after take-off engine speed oil flow was started. Fig. 29 shows the decongealed annulus in the frozen make-up oil provided after 30 min total running time. It has also been determined that with the double-walled hopper weight is saved, and the cost of manufacture is less than for a finned hopper. The finless hopper and diverter-segregator valve with the hopper filler tube and adaptor sump are known as the de-

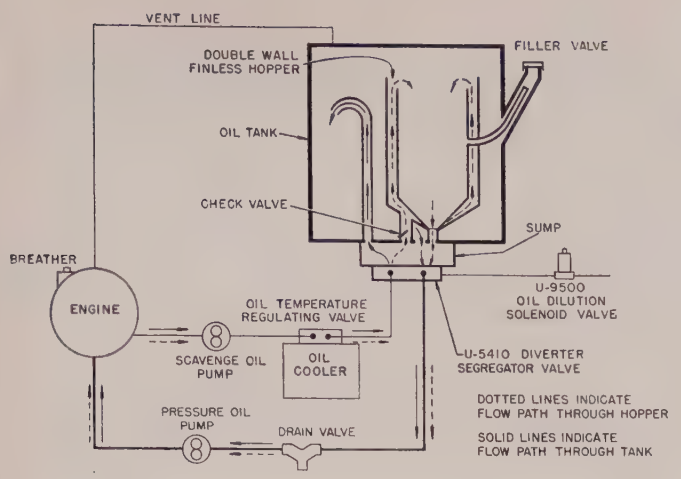

Fig. 26 – Deaerating cold-weather oil system

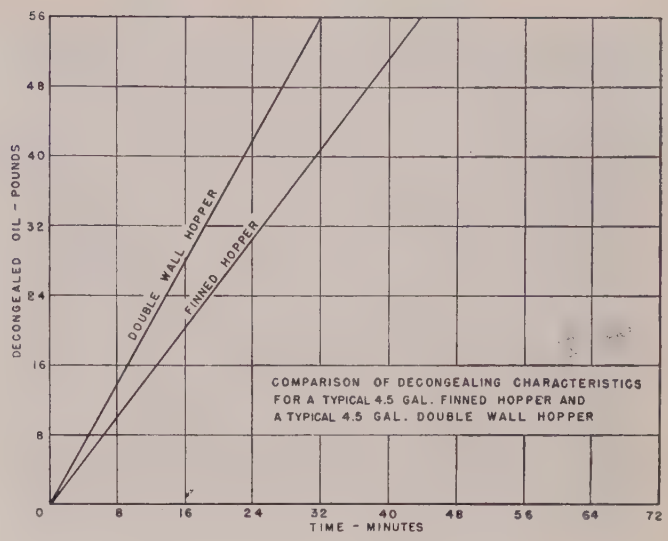
DECONGEALED OIL - POUNDS
TIME - MINUTES
DOUBLE WALL HOPPER
FINNED HOPPER
COMPARISON OF DECONGEALING CHARACTERISTICS FOR A TYPICAL 4.5 GAL. FINNED HOPPER AND A TYPICAL 4.5 GAL. DOUBLE WALL HOPPER

Fig. 27 – Comparison of decongealing characteristics for a typical 4.5-gal finned hopper and a typical 4.5-gal double-wall hopper

aerating cold-weather oil system, Fig. 26. The Beech T-36 oil tank with the deaerating cold-weather oil system is shown in Fig. 30.

Summary

The incorporation of the deaerating cold-weather oil system for the first time will result in oil systems meeting the requirements of the USAF Manual 80-1, "Handbook of Instructions for Aircraft Designers," Par. 14.223 (a) Deaeration and (b) Positive Segregation.

The system performance equals the conventional system performance with regard to providing a given weight flow of oil to the engine and provides markedly superior deaeration and cold-temperature performance over the conventional system.

A detailed analysis of the use of the deaerating cold-weather oil system design by aircraft and USAF design engineers has disclosed that the system inherently includes in-flight "fail-safe" provisions.

Positive segregation of the diluted from the undiluted tank oil is provided which eliminates the necessity of applying externally supplied ground oil system preheat for extended periods of time. It is again emphasized that frozen make-up oil is melted in a minimum of time which eliminates the danger of in-flight engine oil starvation.

In answer to the engine-manufacturers' objections to the conventional oil system design which circulates only a small volume of oil through the engine, this new system circulates all the oil contained in the tank through the engine.

The deaerating cold-weather oil system develop-

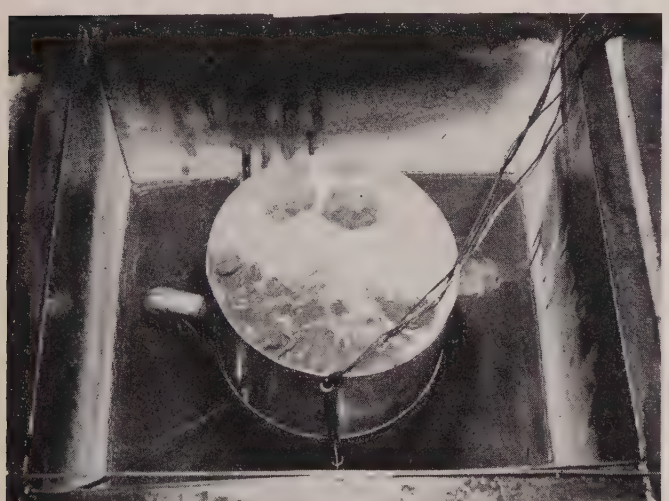

Fig. 28 – Decongealed annulus provided 21 min from start immediately after take-off oil flow was started of —60F test on Beech T-36 deaerating cold-weather oil system

Fig. 29 – Decongealed annulus provided 30 min from start of —60 F test on Beech T-36 deaerating cold-weather oil system

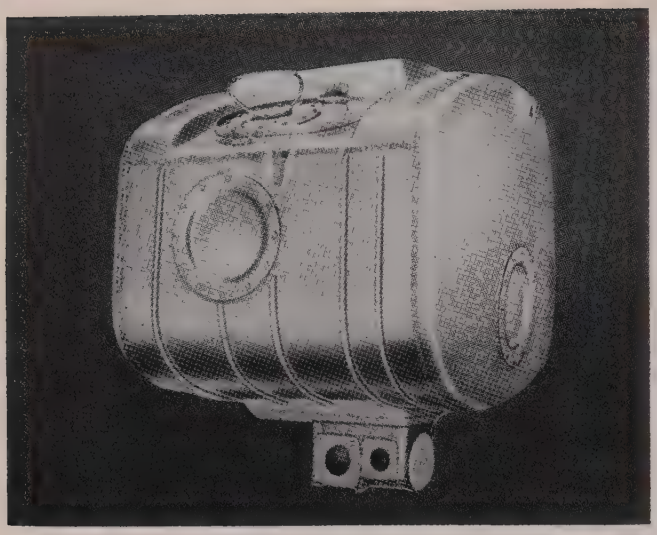

Fig. 30 – Beech T-36 aluminum 32-gal oil tank with UAP deaerating cold-weather oil system

ment has been completed through the combined efforts and/or cooperation of the USAF, RCAF, and aircraft and engine manufacturers. The system answers the many papers written concerning (a) aeration problems and (b) cold-temperature starting problems.

However, the problem of sucessfully starting reciprocating-engine aircraft at cold temperatures has long ago fallen into the same category as Mark Twain's remark about the weather, "Everybody talks about the weather, but nobody does anything about it." As a result, it is understandable that aircraft engineers and operating personnel have come to accept the tremendous effort in terms of ground equipment and personnel required for extended periods of time to prepare an aircraft for an attempted engine start.

But in this case, something has been done about the cold-weather oil-system problem – the deaerating cold-weather oil system is fully developed. The herculean effort formerly required for cold-temperature engine starts can now be eliminated (Fig. 31).

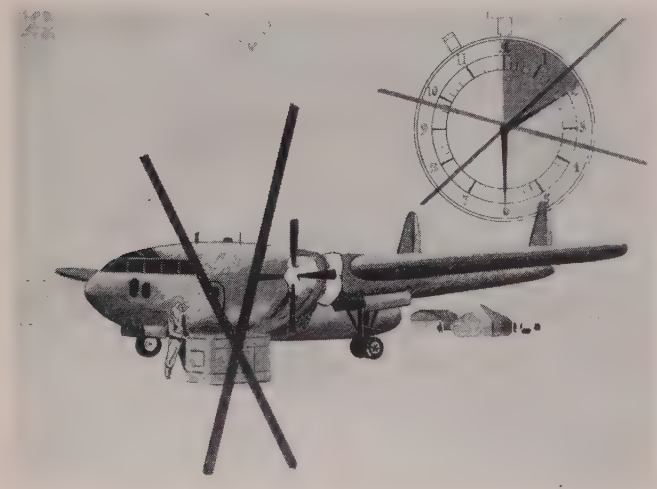

Fig. 31 – Deaerating cold-weather oil system eliminates ground heating equipment and time required to heat oil system

The elimination of the time required for applying ground preheat to oil systems is destined to save untold costs required for ground equipment and personnel and, more important, provide the answer to the military of how to prepare Arctic aircraft for operation "Scramble" – airborne in minutes instead of hours.

Additional References

1. *SAE Journal,* Vol. 53, No. 7, July, 1945, pp. 373-376: "Engine Oil Foaming," by H. A. Ambrose and C. E. Trautman.

2. *Industrial Aviation,* May, 1945, pp. 44-46, 48, 50, 89: "Design of Aircraft Lubrication Systems," by C. S. Brandt.

3. *Royal Aeronautical Society Journal,* Vol. 53, April, 1949, pp. 383-404: "Effect of Aeration of Oil on Behaviour of Engine Lubricating System Under High-Altitude Conditions," by E. Giffen and H. R. Mills.

4. *SAE Journal,* Vol. 51, No. 11, November, 1943, pp. 394-396: "Aircraft Oil Systems – High-Altitude Problems," by H. E. Moerman.

5. ASME Transactions, Vol. 67, July, 1945, pp. 399-404: "Instrument for Indicating Amount of Gas in Gas-Liquid Mixtures," by B. R. Walsh and G. S. Peterson.

6. *SAE Journal,* Vol. 53, No. 7, July, 1945, pp. 402-409: "Oil System Problems at High Altitude," by W. L. Wheeler.

7. Wright Aeronautical Division Field Engineering Memo L-4, Sept. 13, 1948, "Criteria Effecting Optimum Oil System Design for Minimum Air Entrainment."

8. Air Matériel Command Memorandum Report ENG-57-534-31, April 4, 1944, "Oil Systems," by W. Weitzen.

9. Air Matériel Command Memorandum Report MCREXP-534-145, Oct. 14, 1949, "Segregator Oil Tank," by Saul Barron.

10. "Engineering Developments Required for Satisfactory Low-Temperature Aircraft-Engine Operations," by W. Weitzen. Presented at SAE Summer Meeting, French Lick, Ind., June 10, 1948.

11. SAE *Quarterly* Transactions, Vol. 1, April, 1947, pp. 279-289: "Aircraft-Engine Starting Tests and Experiences in the Arctic," by W. Weitzen and R. G. Dunn.

12. USAF APG Final Report No. CW-3-43-2, July 20, 1944, "Cold-Weather Test of Manual Oil Dilution Systems."

13. Air Matériel Command Memorandum Report TSEOC-59, May 22, 1947, "Report on Oil Dilution System Tests Conducted during Winter of 1946-1947 at Ladd Air Force Base, Alaska."

14. ANA Bulletin No. 275d, June 22, 1950, "Guide for Lubrication of Aircraft."

Engine Knock as Influenced by Precombustion Reactions

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Paper was presented at SAE National Fuels and Lubricants Meeting, Chicago, Nov. 6, 1953.

THE influence of fuels, engine design, and operating conditions on knock in internal-combustion engines has been studied extensively for many years. An important result of such research is the conclusion that reaction products generated in the fuel-air charge ahead of the normal flame front are largely responsible for knock in spark-ignition engines. In these investigations a number of different techniques have been used to study the process by which these precombustion or preflame reactions are produced. Some of the more important techniques are:

(a) Pressure-time records of the engine cycle in both motored and spark-ignited engines.

(b) Cool-flame detection in motored engines.

(c) Chemical analysis of the mixture leaving motored engines.

(d) Temperature measurement of the mixture leaving motored engines.

(e) Chemical analysis of the mixture ahead of the flame front in spark-ignited engines.

For fuels which exhibit precombustion reactions, thermodynamic analyses of pressure-time records show that the energy liberated by the reaction may be a significant portion of the total heat of combustion.[1, 2] A relationship between the heat of this reaction and the knocking tendency of fuels has not, however, been established. There is now a great deal of evidence which shows that cool flames in engines occur at the time of heat release during the precombustion reaction period.[3] These cool flames have been identified as light emitted by excited formaldehyde.[4] The possibility that this reaction product – formaldehyde – is the precursor to knock is discounted since it is a relatively poor knock inducer.[5] Sampling and chemical analyses of the products in motored and spark-ignited engines prior to complete combustion of the fuel have shown the presence of aldehydes, peroxides, alcohols, water, and cracked hydrocarbon products.[6, 7, 8] Current theories favor the idea that some peroxide compound plays an important role in the knocking mechanism in engines.

All of this, however, is quite speculative, and none of the investigations have established a direct relationship between the precombustion reaction products generated in the engine combustion chamber and the ensuing knocking reaction. The new engine technique described in this paper makes it possible to study the knock effect of precombustion reaction products in terms of conventional knock-test units and to establish relationships between fuel reactivity and knock.

A pair of variable-compression-ratio, valve-in-head, single-cylinder engines of 17.6 cu in. displacement was used in this study. The fuel-air charge was inducted through a standard CFR carburetor and intake manifold by the first, or motored, en-

[1] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 472-487: "Early Combustion Reactions in Engine Operation," by E. B. Rifkin, C. Walcutt, and G. W. Betker, Jr.

[2] See *Industrial & Engineering Chemistry*, Vol. 43, December, 1951, pp. 2844-2849: "Precombustion Reactions in an Engine," by C. Walcutt and E. B. Rifkin.

[3] See *Industrial & Engineering Chemistry*, Vol. 43, December, 1951, pp. 2805-2814: "Two-Stage Autoignition of Some Hydrocarbons," by W. J. Levedahl and F. L. Howard.

[4] See *Analytical Chemistry*, Vol. 24, 1952, pp. 1776-1780: "Emission Spectra on Autoignited Heptane-Air Mixtures," by W. J. Levedahl and H. P. Broida.

[5] See Philosophical Transactions of Royal Society of London, Series A. Vol. 243, 1951, pp. 299-358: "Reactions that Lead to Knock in Spark-Ignition Engine," by D. Downs, A. D. Walsh, and R. W. Wheeler.

[6] See *Fuel*, Vol. 32, July, 1953, pp. 279-309: "Cool-Flame Formation in a Motored Engine," by D. Downs, J. C. Street, and R. W. Wheeler.

[7] "Preflame Oxidation of Hydrocarbons in a Motored Engine," by A. J. Pahnke, P. M. Cohen, and B. M. Sturgis. Paper presented before Division of Petroleum Chemistry, 124th National Meeting of American Chemical Society, Chicago, Sept. 10, 1953.

[8] "Investigation of Carbonyl Intermediates in Preflame Reactions in an Engine," by E. W. Malmberg, J. A. Bobbitt, F. Zamnik, M. L. Smith, and W. C. David. Paper presented before Division of Organic Chemistry, 124th National Meeting of American Chemical Society, Chicago, Sept. 7, 1953.

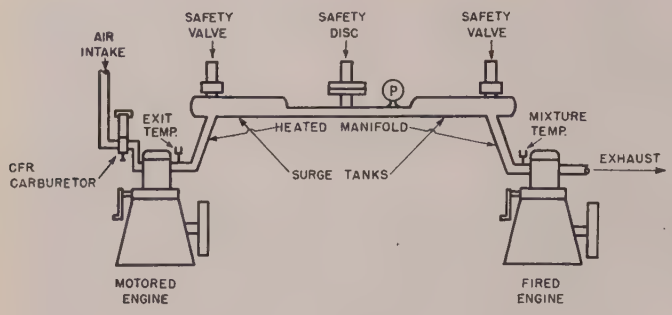

Fig. 1 – Schematic diagram of apparatus

gine; the charge was compressed but not fired, and then expanded and exhausted into a manifold leading to the intake port of the second, or fired, engine. The motored engine was driven by a variable-speed motor while the fired engine was connected to a constant-speed synchronous motor. Thus the two electric motors were completely independent of each other. Fig. 1 is a diagram of the apparatus. The fired engine was operated at 900 rpm, with 145 to 150 F mixture temperature, 212 F jacket temperature, and maximum-power spark advance. The motored engine was operated at a speed of 780 ± 10 rpm, the speed required to maintain a pressure of 29.3 in. of Hg (the average barometric pressure at Detroit) in the manifold between the two engines.

The air supplied to the first engine passed through an ice tower, which maintained the air at a constant humidity of approximately 27 grains of water per pound of dry air. The air was then heated to 150 F – the intake air temperature normally used in these experiments. The fuel/air ratio

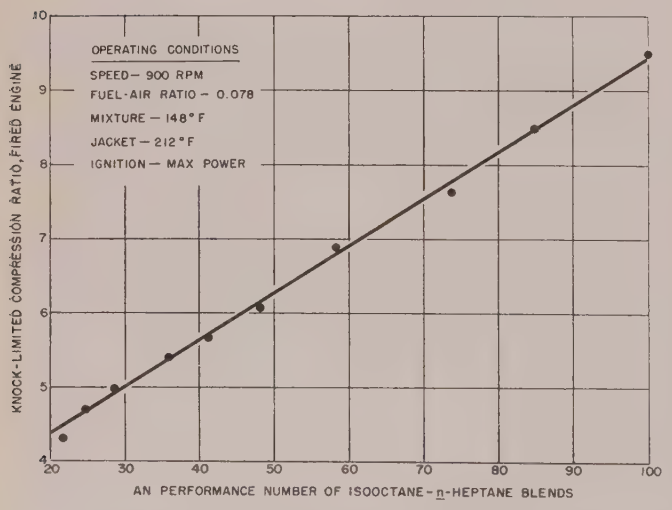
OPERATING CONDITIONS
SPEED – 900 RPM
FUEL-AIR RATIO – 0.078
MIXTURE – 148° F
JACKET – 212° F
IGNITION – MAX POWER
KNOCK-LIMITED COMPRESSION RATIO, FIRED ENGINE
AN PERFORMANCE NUMBER OF ISOOCTANE-n-HEPTANE BLENDS

Fig. 2 – Relationship of fuel performance number and knock-limited compression ratio

was held at 0.077 to 0.078 as determined by volumetric measurements of the fuel and air inducted into the first engine. The manifold connecting the two engines was approximately 15 ft long, and the computed time required for one complete change of gas in the manifold was about 23 sec. This large volume of the connecting manifold was provided as a surge chamber to minimize pulsations. The manifold walls were heated externally in order to maintain a mixture temperature throughout the system of approximately 150 F.

The knocking tendency of the fuel-air charge leaving the motored engine was measured in the fired engine in terms of the compression ratio which resulted in audible trace knock. Frequent calibration of the fired engine over a period of months showed that the knock-limited compression ratio of primary reference fuels could be reproduced within ±0.05 compression ratio.

The antiknock quality of the mixture leaving the motored engine was expressed in terms of the Army-Navy performance number scale. For the conditions used in this work an almost linear relation existed between knock-limited compression ratio and the performance number equivalents of the primary reference fuel blends. The data shown in Fig. 2 were obtained with the motored engine operated at a compression ratio sufficiently low so that the fuels were undergoing no precombustion reactions. This curve was used for each fuel tested to determine the performance number from the observed knock-limited compression ratio.

With the motored engine operating at a compression ratio such that it was, in effect, nothing more than a pump delivering fuel and air to the second engine, knock-ratings made in the second engine with both clear and leaded fuels were slightly higher than their ASTM Research Method ratings.

The technique employed in investigating fuel behavior was as follows: For each fuel the motored engine was operated at a compression ratio of about 5/1 and a measurement made of the knocking tendency of the exit gases in the fired engine. As the compression ratio of the motored engine was increased and repeated knock measurements were made in the fired engine, a point was reached for most fuels at which the antiknock value of the charge from the motored engine decreased. Knock determinations were made at further small increases in compression ratio until the charge autoignited in the motored engine. For some fuels, the antiknock quality of the charge in the fired engine was not affected by the compression ratio of the motored engine, which was limited to a value of 17.25/1 by mechanical interference of the piston and valves. The temperature of the charge leaving the motored engine was measured by a thermocouple located in the exhaust port.

The fuels studied in this investigation included both clear and leaded primary reference fuels and a number of relatively pure hydrocarbons from

each of the four major classes – paraffins, naphthenes, olefins, and aromatics.

The results obtained with isooctane are shown in Fig. 3. As the compression ratio of the motored engine was increased to 11.6/1, isooctane continued to have a performance number of 100, showing that at these compression ratios of the motored engine there were no reactions which affected the knock-rating of the fuel. Further increases in compression ratio of the motored engine resulted in a gradual decrease in the antiknock quality of the charge entering the second engine from 100 to 83 performance number at the 12.5/1 compression ratio level. From this point on, further increases in the compression ratio of the motored engine resulted in an increase of the observed antiknock quality of the charge until, at the autoignition-limited compression ratio of 15.8/1, the antiknock quality was 107 performance number, or seven units above that of the unreacted isooctane.

A curve of the temperature of the charge leaving the motored engine as the compression ratio was increased is also shown in Fig. 3. A gradual decrease in temperature with increase in compression ratio was observed up to about 12/1. This decrease is attributed to heat losses during the cycle, which gradually become greater as the peak temperatures of the cycle are increased. Above 12/1, however, a decided increase in the temperature of the charge leaving the motored engine was observed. It is important to note that the compression ratio at which a distinct increase in temperature was observed occurs at the point on the upper curve corresponding to the point on the lower curve where there is a significant change in the rate of decrease in antiknock quality of the charge. This indication of exothermic reaction occurs at a higher compression ratio than that at which the strong knock-inducing reactions were first detected. Thus, at the point at which knock-inducing products were first produced in the motored engine, there was no evidence of any heat liberation, that is, the reaction was thermoneutral. The fact that reactions had occurred was acutely evident, however, by the distinctive odor of the exit gases from the motored engine when it was operated above 11.6/1 compression ratio.

Pressure-time records obtained by means of an instantaneous pressure indicator in the motored engine, when operated over this range of compression ratios, also indicated that no detectable energy was being released below the compression-ratio level at which exothermic reactions were indicated by an increase in temperature of the exit charge. However, at compression ratios where appreciable exothermic reactions were detected, there was a definite pressure indication of energy release. Measurements of the power output of the fired engine also showed a continued decrease as the exothermic reactions increased in magnitude.

The increase in the antiknock quality of the charge following liberation of heat indicates that

A NEW experimental approach for evaluating influence of precombustion reactions on engine knock and some results obtained by this method are described here. Measurements in terms of conventional knock-test units were made of the knocking tendency of a fuel-air mixture previously compressed in a motored engine.

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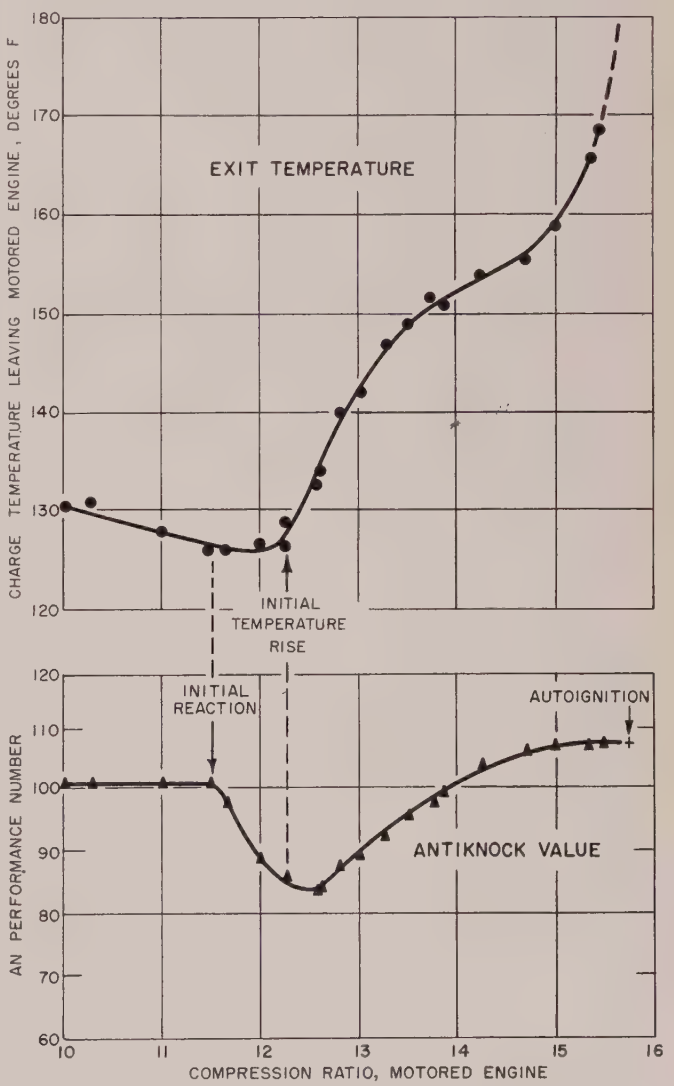

Fig. 3 – Precombustion reactions exhibited by isooctane

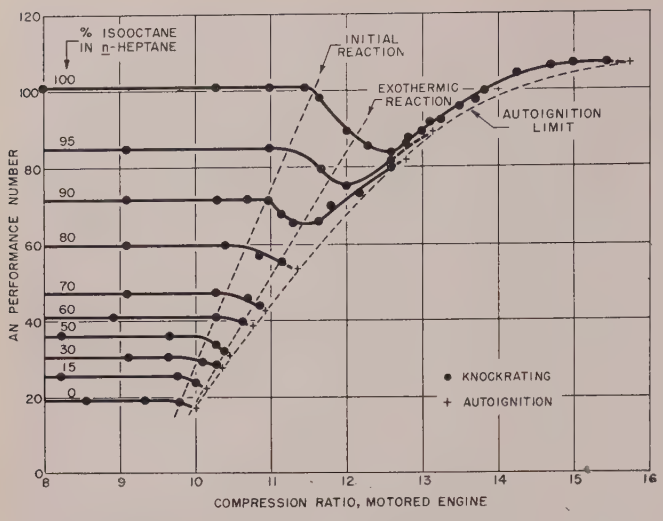

Fig. 4 – Effect of precombustion reactions on antiknock quality of primary reference fuels

the reaction process producing this large-scale heat effect does not produce products which are strongly knock inducing. The knock suppression observed at this time is attributable to a decrease in the heat content of the charge received by the fired engine when extensive exothermic reactions are occurring in the motored engine. The situation is somewhat similar to the effect produced by throttling an engine.

Primary Reference Fuels – Results obtained with blends of n-heptane and isooctane are shown in Fig. 4. Data were obtained for each fuel, as previously described. The dashed line labeled "initial reaction" in the figure was drawn through the point at which a decrease in antiknock quality of each fuel was first observed. The dashed line labeled "exothermic reaction" denotes the compression ratio at which these reactions were detected by the initial increase in charge exit temperature. At the autoignition limit, the values of fuel antiknock quality were determined by a small extrapolation of each curve to the autoignition-limited compression ratio of the motored engine.

It is apparent that all of the primary reference fuel blends undergo some type of precombustion reactivity prior to autoignition and that the extent of these reactions increases with increasing fuel antiknock quality. All blends show the initial or thermoneutral type of reaction discussed previously for isooctane, while blends of high antiknock quality also show extensive exothermic reactions.

Leaded Primary Reference Fuels – Blends of primary reference fuels containing up to 6.0 ml tel per gal were investigated. As expected, the addition of tetraethyl lead increased the antiknock quality of the fuel-air charge leaving the motored engine, regardless of the compression ratio at which the engine was operated. The leaded fuels were similar to the unleaded primary reference fuels in that only fuels of high antiknock value showed both thermoneutral and exothermic types of reactions. Results on four of these primary reference fuel blends, each containing 6.0 ml tel per gal, are plotted in Fig. 5 against the background of results obtained for the unleaded blends of primary reference fuels. A comparison of the data obtained from these leaded blends with those from unleaded blends of the same base composition indicates that the knock-inducing and knock-suppressing reactivity of the leaded blends is more extensive than for the unleaded base. It is evident that the presence of tetraethyl lead in a fuel permits the engine to be operated at an increased compression ratio without autoignition, but the tetraethyl lead in no way retards the greater precombustion reactivity of the base fuel at the increased compression ratio.

Experiments were also conducted with a 60 octane number primary reference fuel containing various concentrations of tetraethyl lead. The data obtained in these tests are shown in Fig. 6, where performance number in the fired engine and charge exit temperature are plotted against the compression ratio of the motored engine. These data clearly indicate that the addition of tel does not affect the compression ratio of the motored engine at which the initial precombustion reactions first occur. Previous studies, by means of pressure-time records,[1] have shown that tetraethyl lead has no effect on the start of exothermic precombustion reactions. This conclusion is supported by the present work. Reference to Fig. 6 shows that three of the leaded blends (60 octane number primary reference fuel containing 3.0, 4.6, and 6.0 ml tel per gal) exhibited an increase in charge exit temperature and, therefore, exothermic precombustion reactivity at the same compression ratio, 11.3/1. In addition, the current investigation establishes the fact that tetraethyl lead has no effect on the start of the thermoneutral reactions which produce proknock materials, as evidenced by the line on the figure labeled "initial reaction." The autoignition-limited

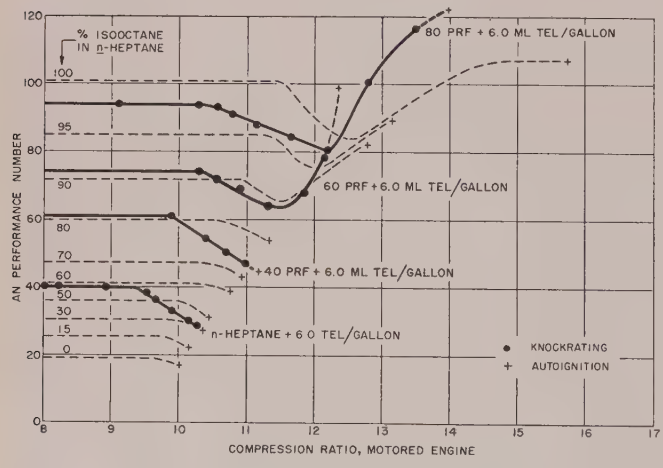
% ISOOCTANE IN n-HEPTANE
80 PRF + 6.0 ML TEL/GALLON
60 PRF + 6.0 ML TEL/GALLON
40 PRF + 6.0 ML TEL/GALLON
n-HEPTANE + 6.0 TEL/GALLON
• KNOCKRATING
+ AUTOIGNITION
AN PERFORMANCE NUMBER
COMPRESSION RATIO, MOTORED ENGINE

Fig. 5 – Effect of precombustion reactions on antiknock quality of leaded primary reference fuels

compression ratio, however, increases progressively as tetraethyl lead concentration is increased.

Since tetraethyl lead does not affect the formation of the proknock materials observed in this investigation but does suppress knock, the inference is drawn that tetraethyl lead interferes with the knock-promoting properties of these reaction products and not with their formation. In other words, if a critical concentration of active centers is necessary for knock, it would appear that tetraethyl lead functions in some intermediate step between the formation of the proknock materials observed in this investigation and the ensuing knock reaction.

Diisobutylene-n-Heptane Blends – Diisobutylene (dib), a mixture of 2, 4, 4-trimethyl-1-pentene and 2, 4, 4-trimethyl-2-pentene, is another reference fuel frequently used in engine tests where fuel sensitivity is a factor. Dib alone and two blends of dib and n-heptane were investigated. The results of these tests are shown in Fig. 7 against a background of the data for the primary reference fuels. When 100% dib was used, no reactions could be detected either by an increase in the knock intensity of the fired engine or by an increase in the temperature of the charge leaving the motored engine – even though the compression ratio of the motored engine was increased to its maximum limit of 17.25/1.

The two blends of 60 and 80% by volume dib in n-heptane, however, exhibited the initial thermoneutral type of precombustion reactions in which there was a steady decrease in antiknock quality with increasing compression ratio of the motored engine. Only a slight increase in charge exit temperature immediately preceding autoignition was observed, in contrast with the marked exothermic reactions obtained with the primary reference fuels. A comparison of these two blends with isooctane-n-heptane blends of either equal antiknock quality or equal n-heptane content shows that dib is more effective than isooctane in improving the resistance of n-heptane blends to thermoneutral precombustion reactivity and to autoignition at the operating conditions used in this investigation. Since in this study the operating conditions of the motored engine were quite mild in that there was no observed reactivity for dib, it appears that under these conditions, undecomposed dib acts as a "sink" for the reaction products generated by n-heptane. In other words, dib has good knock resistance both by itself and as a blending agent, provided the engine operating conditions are sufficiently mild that the dib is undecomposed. At more severe engine operating conditions, the dib might be decomposed and lose its knock-inhibition effect for more reactive fuel types like n-heptane.

Other Hydrocarbons – The results obtained for several hydrocarbons of each of the following types – olefins, naphthenes, and paraffins – are shown in Fig. 8 against the primary reference fuel background. The effect of fuel type may be summarized as follows:

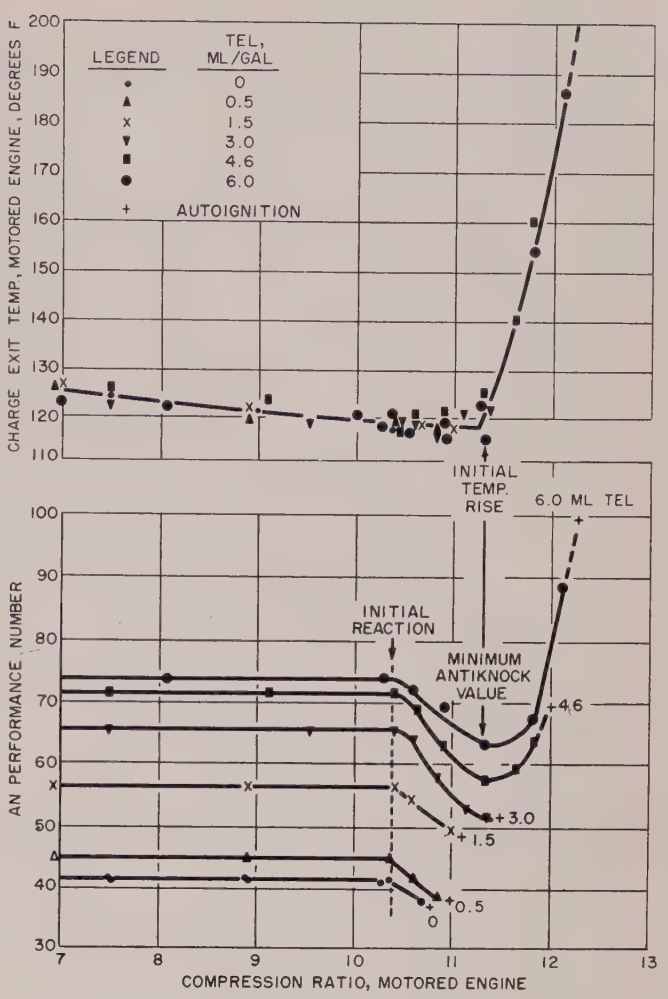

Fig. 6 – Effect of tetraethyl lead on precombustion reactions exhibited by a 60 octane number primary reference fuel

1. Two aromatics, benzene and toluene, could not be made to knock in the fired engine, and no precombustion reactions could be detected by measurements of the temperature of the fuel-air charge

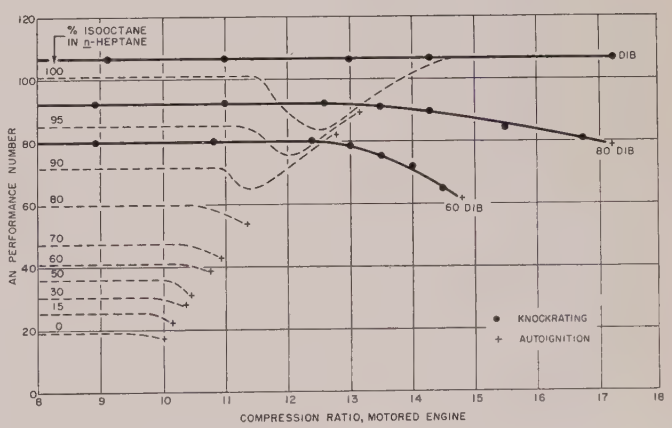

Fig. 7 – Effect of precombustion reactions on antiknock quality of diisobutylene blends

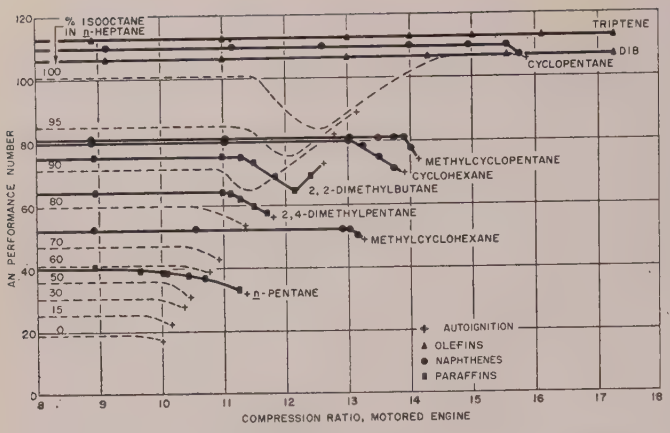

Fig. 8 – Effect of precombustion reactions on antiknock quality of various hydrocarbons

leaving the motored engine. Therefore, they cannot be shown on the plot of Fig. 8.

2. Two olefins, triptene and dib, did knock in the fired engine but could not be made to react in the motored engine and showed no indications of reactivity in the motored engine under these conditions.

3. Four naphthenes, cyclopentane, methylcyclopentane, cyclohexane, and methylcyclohexane, ranging from 50 to 115 performance number, could be made to knock in the fired engine and to show knock-inducing reactivity in the motored engine, with no liberation of heat until the compression ratio was raised to the point of autoignition. It can be seen from Fig. 8 that these naphthenes require higher compression ratios for the initiation of precombustion reactions in the motored engine than do the primary reference fuels of equal antiknock quality.

4. The five paraffins, n-heptane, n-pentane, 2, 4-dimethylpentane, 2, 2-dimethylbutane, and isooctane, show the least resistance as a class to the start of the initial knock-inducing precombustion reactions. The compression ratio at which precombustion reactions first occurred in the motored engine was almost linearly related to the performance number of these paraffins. They were the only hydrocarbons to show any increase in antiknock quality prior to the onset of autoignition.

The effects of both fuel/air ratio and carburetor air temperature on precombustion reactivity were studied with two fuels. Because of its known sensitivity to engine operating conditions, a blend was selected of 60% dib plus 40% n-heptane, having an octane number of 93. The other fuel of approximately the same antiknock value but of less sensitivity was a 90 octane number primary reference fuel blend. The data for each blend were obtained within a period of two or three days, so as to minimize effects of changes in barometric pressure and cleanliness of the motored engine.

Carburetor Air Temperature – With the carburetor air temperature varied from 100 to 250 F and the fuel/air ratio maintained at 0.078, the 60% dib blend required a higher compression ratio for the initiation of precombustion reactions in the motored engine than the primary reference fuel blend. Fig. 9 shows that the compression ratios at which the initial reactions take place decrease linearly and to nearly the same degree with both fuels as carburetor air temperature is increased. Thus, the two fuels have about the same sensitivity with regard to initial precombustion reactivity. On the other hand, although the autoignition-limited compression ratio of the dib blend at 100 F carburetor air temperature is appreciably greater than that of the 90 primary reference fuel, as is also shown in Fig. 9, at 250 F the two fuels are autoignition limited at about the same compression ratio. Since dib is considered to be temperature sensitive by normal knock-rating procedures, these results are in keeping with other published results[9] which indicate that knock is an autoignition reaction.

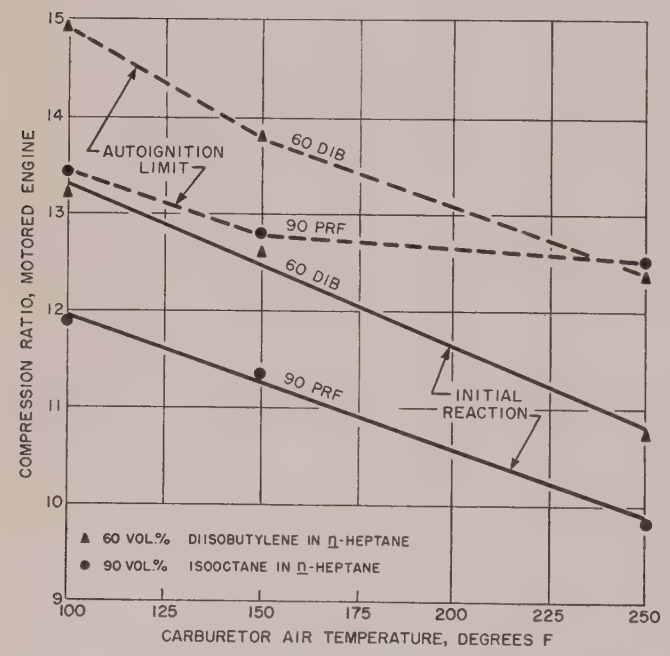

Fig. 9 – Influence of carburetor air temperature and fuel sensitivity on precombustion reactions – fuel/air ratio 0.078

Fuel/Air Ratio – The results obtained during an investigation of the effect of fuel/air ratio are shown in Fig. 10. With the fuel/air ratio varied from 0.050 to 0.094 and the carburetor air temperature maintained at 150 F, the compression ratio at which initial reactions first occur was a minimum at approximately 0.060 fuel/air ratio for both the dib and the primary reference fuel blends. The dib blend required a somewhat higher compression ratio for these initial reactions than the primary reference fuel blend at all fuel/air ratios

[9] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 584-585: discussion by C. Walcutt.

investigated. This difference, however, was much greater at the leaner mixtures. The autoignition-limited compression ratio of the primary reference fuel blend, on the other hand, was little affected by changes in fuel/air ratio, while that of the dib blend varied considerably. With both fuels, the autoignition-limited compression ratio was a minimum at lean mixtures.

Effect of Pressure and Temperature on Precombustion Reactions – The fact that some of the fuels tested, such as n-heptane, exhibited no exothermic reactions preceding autoignition in the motored engine, while others, such as isooctane, did undergo extensive reactivity should not be taken as a fundamental distinction between these two fuels. This difference is due primarily to the particular engine operating conditions used in the investigation. In a particular engine, the extent of the precombustion reactivity, at least in the case of paraffinic-type fuels, depends on the temperature-pressure-time path to which the fuel is subjected. For example, n-heptane, which showed no evidence in these tests of any exothermic precombustion reactions prior to autoignition, has shown extensive exothermic precombustion reactions in a motored engine operating at the autoignition-limited compression ratio[1] as well as in a flame tube.[10, 11]

The calculated temperatures at which the initial precombustion reactions were first detected in the motored engine were little affected by fuel antiknock quality. With the two primary reference fuels, n-heptane and isooctane, for example, initial reactions in the motored engine operating at 150 F carburetor air temperature and 0.078 fuel/air ratio occurred at compression ratios of 9.7/1 for n-heptane and 11.6/1 for isooctane. The measured corresponding peak pressures were 250 and 307 psia, and the computed temperatures were 548 and 572 F respectively. Similar data for other primary reference fuel blends are shown in Fig. 11.

The temperatures were computed by means of the perfect-gas law, measured peak combustion-chamber pressures, and engine volumes. The fact that these values are considerably lower than would be expected indicates that there is considerable heat leak in this engine. Since the engine was operated at high compression ratios and with comparatively cool combustion-chamber walls, these relatively low temperatures seem reasonable.

These data show that there is a difference of only 24 F in the temperatures at which precombustion reactions are initiated when the motored engine was operated on n-heptane and isooctane. However, it should be borne in mind that there was also a difference in peak pressure of 57 psi between

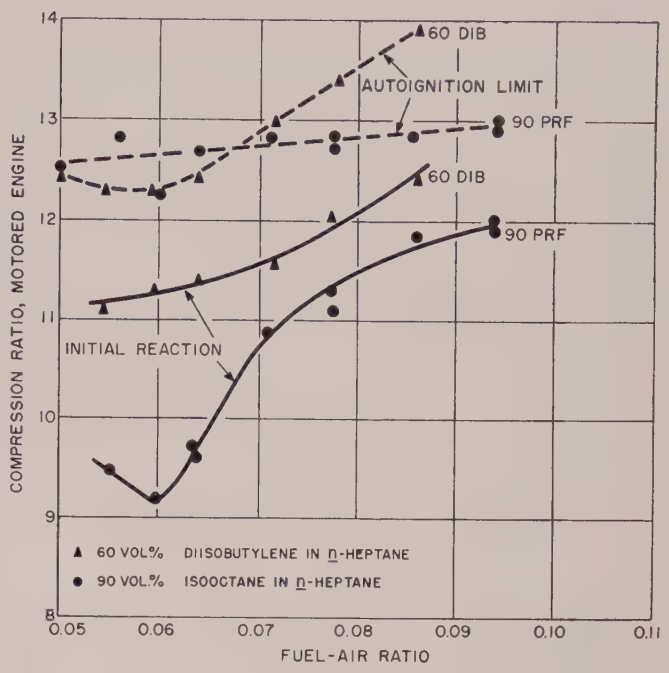

Fig. 10 – Influence of fuel/air ratio and fuel sensitivity on precombustion reactions – carburetor air temperature 150 F

these two fuels at the point of initial reaction. Research currently under way in the Ethyl Laboratories on the pressure-temperature relationship during knocking in engines shows that in some cases pressure may be of even greater importance than temperature.

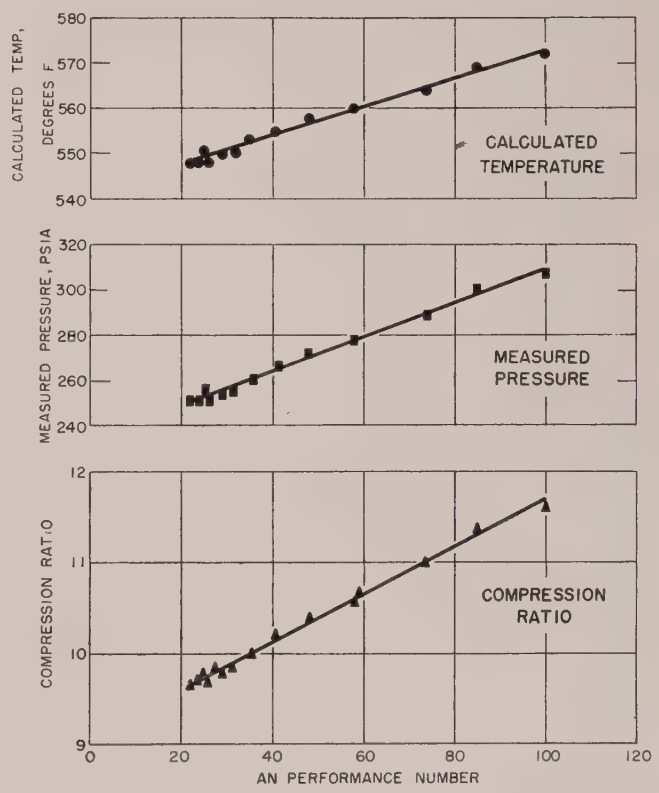

Fig. 11 – Relationship of fuel antiknock quality and engine conditions required for precombustion reactions

[10] See SAE Transactions, Vol. 61, 1953, pp. 398-399: Discussion by P. E. Oberdorfer.

[11] "Effect of Preflame Oxidation Reactions on Engine Knock," by C. Walcutt, J. M. Mason, and E. B. Rifkin. Paper presented before Division of Petroleum Chemistry, 124th National Meeting of American Chemical Society, Chicago, Sept. 10, 1953.

The utilization of a new engine technique to study the influence of precombustion reactions on the knocking tendency of fuels has been demonstrated. When a normally operated engine is run on a fuel-air mixture that is previously compressed but not fired, the change in antiknock quality of the mixture is an extremely sensitive indication of the precombustion reactions preceding normal combustion. Of particular importance in this respect is the fact that the thermoneutral reactions occurring in the fuel-air mixture have been shown to produce strong proknock materials. Precombustion products from exothermic reactions occurring in the mixture, however, are not strongly knock inducing.

The addition of tetraethyl lead to a fuel does not influence the compression ratio at which either the thermoneutral or exothermic precombustion reactions occur, but does act to retard the onset of autoignition. It is concluded that tetraethyl lead suppresses knock by controlling, in some manner, the decomposition products generated from stable intermediates, but it has no influence on the reaction process by which these intermediates are produced.

The aromatics and olefins tested show no precombustion reactivity under the operating conditions used in this investigation. The naphthenes tested show only thermoneutral reactions and are more resistant in this respect than the paraffins of equal antiknock quality. The paraffins show the least resistance to the thermoneutral precombustion reactivity and are the only class of hydrocarbons to show extensive exothermic reactions.

Increases in carburetor air temperature decrease the compression ratio at which thermoneutral reactions start, both with a dib-heptane and an isooctane-heptane blend. The autoignition-limited compression ratio for the dib blend decreases more with increasing carburetor air temperature than the isooctane blend. This temperature sensitivity for dib is in agreement with results obtained by conventional knock-rating procedures. The minimum compression ratio both for initial thermoneutral reactivity and for autoignition with these two blends occurs at approximately 0.060 fuel/air ratio.

The investigation indicates that the thermoneutral reactions which produce the active proknock materials may be influenced to a considerable degree by pressure as well as by temperature in the combustion chamber.

DISCUSSION

Cool-Flame Occurrence Shown by Peroxide Content

– J. C. Firey
California Research Corp.

THE authors are to be complimented on the presentation of an excellent paper describing an altogether new approach to the old problem of knock. The two-stage or perhaps multistage nature of the knocking process has been recognized for some time, but attempts to separate the stages and investigate both independently have not been altogether successful until recently. In tubes and bombs, the cool-flame processes have been often examined, but only recently has the subsequent "autoignition" or knock process also been obtained in tubes.[a] The present authors have succeeded in separating the processes in actual engines, and this new technique promises to tell us much about knock in engines. Were the authors able to ascertain the degree of separation achieved, or specifically did secondary cool-flame reactions occur in the fired engine with or without the previous occurrence of a cool flame in the motored engine? Double or multiple cool flames have been observed in tubes and engines.[b]

The authors show fuel antiknock value in the fired engine to drop prior to the occurrence of a cool flame in the motored engine and to recover when cool flames are obtained. If we blame organic peroxides for relative loss of antiknock value, these results are remarkably parallel to those obtained by Pahnke, Cohen, and Sturgis.[c] These investigators found the organic peroxide content of a motored engine exhaust to increase prior to occurrence of a cool flame and to decrease when a cool flame is obtained.

Precombustion reactions certainly appear to produce knock as an end product; but, in the process, the mixture which is ultimately to knock has been drastically changed from its original form. These changes appear to be related to gasoline sensitivity, as indicated by the inverse relation between gasoline sensitivity (that is, F-1 minus F-2 rating) and cool-flame intensity in a motored engine shown by Downs, Street, and Wheeler.[b] The authors show a directionally similar effect in the relative absence of cool-flame reactivity of blends containing the "sensitive" fuel, diisobutylene. A good deal of the apparent complexity of gasoline sensitivity may well result from the fact that it is measured in terms of primary reference fuels which, as the authors demonstrate, undergo appreciable precombustion reactions.

[a] See "Fourth Symposium (International) on Combustion"; pp. 396-402: "Some Observations by Schlieren Technique of Propagation of Flames in a Closed Vessel," by A. C. Egerton, O. A. Saunders, A. H. LeFebvre, and N. P. W. Moore. Pub. by Williams & Wilkins, Baltimore, 1953.

[b] Same as footnote No. 6 of main paper.

[c] Same as footnote No. 7 of main paper.

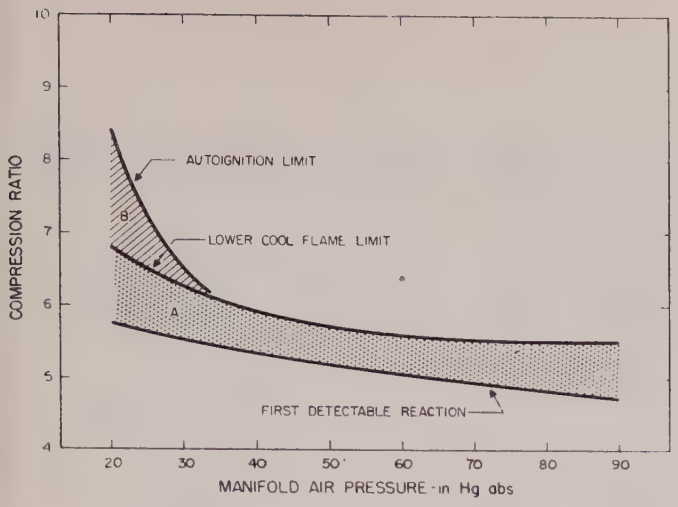

Fig. A – Precombustion reaction limits

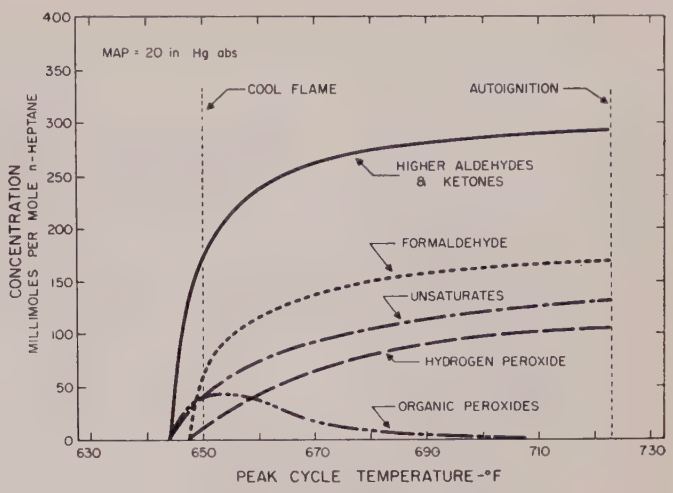

Fig. B – Exhaust-gas composition – n-heptane

Gives Data Concerning Precombustion Reaction Products

– D. L. Pastell

E. I. du Pont de Nemours & Co.

MESSRS. Mason and Hesselberg are to be commended for their very fine contribution to the rapidly growing knowledge of the physical and chemical processes leading to engine knock. This work is of particular interest because of the novel approach taken by the authors. It is only through the use of new experimental approaches that a complete knowledge of the knocking process can be attained. Chemical studies of the reactions resulting in knock carried out at the du Pont Petroleum Laboratory confirm many of the conclusions reached by the authors and supplement their explanations of the significance of precombustion reactions in engines.

The authors attributed their failure to detect exothermic reactions in n-heptane to the particular temperature-pressure-time path associated with their engine operating conditions. That this is probably the correct explanation is shown in Fig. A which defines the initial peroxidation, cool-flame, and autoignition limits for a stoichiometric n-heptane-air mixture over a wide range of engine operating conditions. Previously published data[d] have shown that cool-flame initiation marks the beginning of exothermic reactions. Region A in the figure, the zone of peroxidation preceding cool-flame formation, is the region of thermoneutral reactions and region B – the cool-flame region – the zone of exothermic reactions. With the engine and operating conditions used to develop the data of Fig. A, the apparent intersection of the cool-flame and autoignition limit curves occurs at a manifold air pressure of about 34 in. of Hg abs, or at very nearly the same manifold pressure employed by the authors. Actually, autoignition is preceded by cool-flame formation even after the cool-flame and autoignition curves appear to become a single curve; the cool-flame reactions in this region proceed at such a high rate that autoignition follows very rapidly after the initial cool-flame formation. While the data shown in Fig. A were obtained at an engine speed of 1800 rpm and the authors' data were developed at 780 rpm, the effect of a reduction in engine speed is to shift the intersection of the cool flame and autoignition limits to lower manifold air pressures.[e] Thus, the engine conditions employed by Mason and Hesselberg represent a point following the cool-flame and autoignition curve intersection, and no measurable exothermic reaction was detected.

When the engine conditions are chosen to follow a temperature-pressure-time path resulting in high exothermicity, it is possible to sample and analyze quantitatively the precombustion reaction products starting from the point of the initial peroxidation up to incipient autoignition. For example, when using a manifold air pressure of 20 in. of Hg abs, the initial reactions are first detected at a compression ratio of 5.75/1. Samples of the reaction products for chemical analyses are secured at this compression ratio and at successively higher compression ratios up to the point of autoignition. The results[e] of such analyses are shown in Fig. B for a stoichiometric n-heptane-air mixture. The two-stage character of the n-heptane oxidation can be seen from the composition curves. The first stage is represented by the initial formation and rapid buildup of organic peroxides, unsaturates, higher carbonyl compounds, as well as formaldehyde and hydrogen peroxide. This first stage is the thermoneutral zone observed by the authors. Throughout the second stage the buildup of these products occurs at a much slower rate with the exception of the

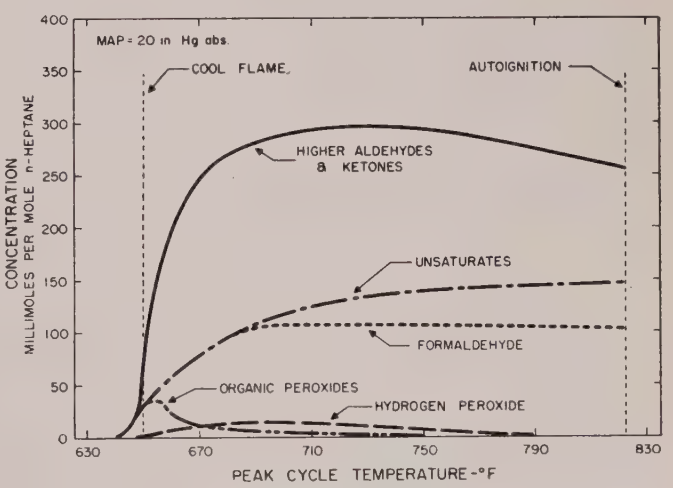

Fig. C – Exhaust-gas composition – n-heptane plus 3 ml tel per gal

[d] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 571-587: "Precombustion Reactions in a Motored Engine," by D. L. Pastell.

[e] See SAE Transactions, Vol. 61, 1953, pp. 386-401: "Some Factors Affecting Precombustion Reactions in Engines," by M. W. Corzilius, D. R. Diggs, and D. L. Pastell. (Refer particularly to p. 394.)

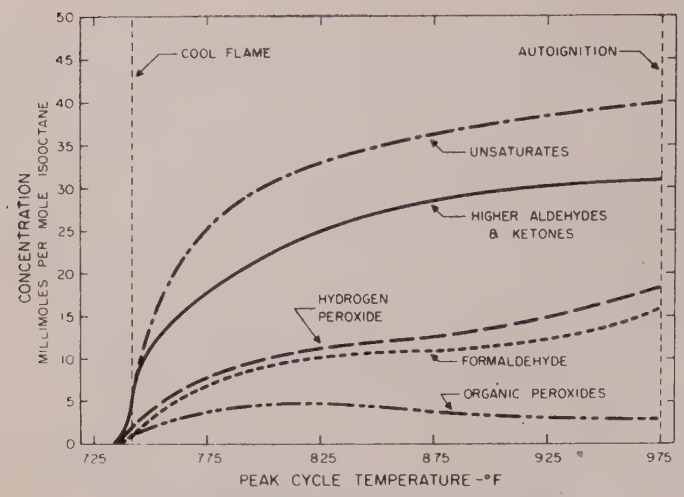

Fig. D – Exhaust-gas composition – isooctane

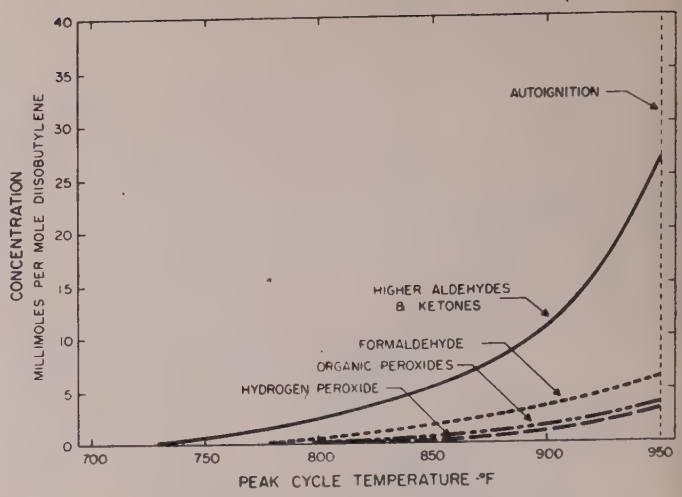

Fig. E – Exhaust-gas composition – diisobutylene

organic peroxides, which decrease to trace amounts at the point of autoignition. During this second stage an appreciable conversion to water, carbon monoxide, and carbon dioxide is found in addition to the intermediates listed. The formation of these latter products involves extremely exothermic reactions and represents a substantial quantity of heat release. A material balance has shown that more than 70% of the n-heptane has reacted prior to autoignition. The ultimate autoignition process is, therefore, one involving less than 30% of the original n-heptane and considerable quantities of oxygenated substances.

The effect on the concentration of precombustion reaction products due to the addition of a 3 ml per gal of tetraethyl lead to n-heptane is shown in Fig. C. No change in the temperature required for initial reaction or for cool-flame formation results from the tetraethyl lead addition. The most pronounced effect of tetraethyl lead is to lower the concentration of formaldehyde and almost completely eliminate the hydrogen peroxide. This antiknock agent has little effect on the organic peroxides, higher carbonyl, and unsaturate concentrations indicating, as suggested by Mason and Hesselberg, that the reactions leading to the formation of these intermediates are not influenced by the presence of tetraethyl lead.

The reduction in the concentrations of hydrogen peroxide and formaldehyde is the most significant change brought about by the addition of tetraethyl lead. Since the radicals from the decomposition of organic peroxides are believed to be the precursors of hydrogen peroxide and formaldehyde, this effect would indicate that tetraethyl lead destroys these radicals. Chamberlain, Hoare, and Walsh[f, g] have presented data indicating that the active specie derived from tetraethyl lead exerts antiknock action through deactivation of the HO_2 radical. Since it is believed that hydrogen peroxide can be formed by the collision of two HO_2 radicals,[h] the reduction in the hydrogen peroxide concentration in the precombustion reaction products of n-heptane due to the addition of tetraethyl lead can be explained by the destruction of the HO_2 radicals by the antiknock.

The buildup of various intermediates during the precombustion reactions of isooctane shows the same general trends observed for n-heptane. The precombustion reaction product curves for isooctane are shown in Fig. D. These data were obtained at a higher engine manifold air pressure than the data shown for n-heptane. Even under the more severe pressure conditions, the initial peroxidation of isooctane was found to occur 81 F higher than for n-heptane. An examination of the individual curves shows that the peroxides formed by isooctane still exist at temperatures where the organic peroxides formed by n-heptane have completely decomposed, indicating the greater stability of the isooctane peroxides. Another difference in the behavior of the two hydrocarbons is the change in the ratio of formaldehyde to hydrogen peroxide from 2/1 for n-heptane to approximately 1/1 for isooctane.

The degree of oxidation of isooctane at incipient autoignition is only about 10% as much as that for n-heptane. This can be ascribed to the greater difficulty of initial oxygen attack on the branched structure of the isooctane molecule and to a greater stability of the products formed.

The authors have correctly stated that diisobutylene does not exhibit physical manifestations of precombustion reactions. However, as shown in Fig. E, it must not be assumed that this fuel does not undergo chemical reaction prior to autoignition. The oxidation of this hydrocarbon, which is quite different from that previously illustrated for n-heptane and isooctane, proceeds very slowly with increase in temperature and does not give rise to cool-flame formation. Apparently diisobutylene exerts a self-inhibition effect due to the activity of the double bond in deactivating the radicals necessary for continued attack on unreacted diisobutylene. These results explain the excellent blending characteristics of this hydrocarbon, for it is reasonable to suppose that diisobutylene not only inhibits its own oxidation but likewise the oxidation of other hydrocarbons with which it is blended.

The concentration curve for the organic peroxides shows these products to be quite stable. The absence of cool flames with this hydrocarbon can be attributed to this factor, since cool flames are believed to result from reactions of organic peroxide decomposition products.[e]

One final comment should be made concerning the technique employed by authors. The very long residence time (23 sec) of the precombustion reaction products in the manifold connecting the two engines would be expected to result in the decomposition of unstable intermediates such as peroxides. Therefore, the proknock effects ascribed to peroxidic materials formed in the thermoneutral zone may well be much greater than shown by the data of Mason and Hesselberg. This in no way invalidates the conclusions reached by the authors but rather points out the difficulties that may be encountered in a study of the reactions leading to knock.

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[f] See Proceedings of Royal Society of London, Series A, Vol. 215, November, 1952, pp. 175-186: "Inhibiting Effect of Lead Tetraethyl. I. Effect of Lead Compounds on Vapor-Phase Slow Oxidation of Diisopropyl Ether and on Ignition of Diethyl Ether," by G. H. N. Chamberlain and A. D. Walsh.

[g] See Proceedings of Royal Society of London, Series A. Vol. 215, December, 1952, pp. 454-466: "Inhibiting Effect of Lead Tetraethyl. II. Effect of Lead Monoxide on Slow Oxidation of Methane," by D. E. Hoare and A. D. Walsh.

[h] See p. 106 of "Combustion, Flames, and Explosions," by B. Lewis and G. von Elbe. Pub. by Academic Press, Inc., New York, 1951.

Measurement and Evaluation of Exhaust Noise of Over-the-Road Trucks

D. B. Callaway, formerly with Armour Research Foundation

This paper was presented at the SAE National Transportation Meeting, Chicago, Nov. 3, 1953.

THE measurement and control of exhaust noise of highway trucks and other vehicles has for many years been a problem of interest and concern to the SAE. The most recent report to the SAE on this subject was made in October, 1950, by Dr. Paul Huber, who was a member of the Traffic Noise Subcommittee of the SAE Truck and Bus Committee.[1] According to that report, the committee concluded among other things:

1. That noise meters complying with ASA Standard Z24.3-1944 gave test results which showed extremely poor correlation with loudness and annoyance judgment tests by human observers; and

2. That a special meter having three channels or frequency bands driven by a single microphone would probably give more satisfactory results.

This committee did a thorough job of preparing specifications for a three-band meter. With the outbreak of the Korean war, however, it was found impossible to interest instrumentation manufacturers in designing the desired equipment, and estimates as high as $50,000 were received for further developmental work.

In early 1952, the American Trucking Associations became interested in the problem and held preliminary discussions with Armour Research Foundation to determine what our approach to the problem would be. As a result, a one-year research program was set up having the following objectives:

1. To develop a practical method of measuring truck exhaust noise which utilized commercially available instrumentation and which gave results substantially in agreement with loudness judgments by qualified listeners.

2. To obtain statistical data on the range of loudness of truck noise for all types of over-the-road trucks in different parts of the country.

3. To show that test-stand noise measurements could be correlated with the noise measurements made on the highway.

This program has now been completed, and technical reports covering the work are in the hands of ATA. The purposes of this paper are to review the work which has been accomplished, to summarize the present status of technical aspects of

RESULTS of an Armour Research Foundation program, prompted by ATA interest in truck exhaust noise problems, are described here.

It is shown that exhaust-noise measurements can be made using an octave band analyzer, and values thus obtained agree quite well with listening tests. Highway truck noises are recorded on magnetic tape, which is then fed into the analyzer at leisure.

Test-stand measurements, however, give good results only when the exhaust is the main source of truck noise.

The Author

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[1] See SAE *Quarterly* Transactions, Vol. 5, October, 1951, pp. 532-543: "Report of Automotive Traffic Noise Subcommittee," by Paul Huber.

the problem, and to suggest some areas in which further research should be done in order to bring about an improvement in the truck noise problem.

It has been recognized for several years that the American Standard Sound Level Meter does not give results which accurately represent the loudness of complex noises, even when the recommended weighting networks provided with the instrument are used. Recent developments in instrumentation and measurement techniques, however, indicate that use of suitable frequency analyzing equipment in conjunction with the sound level meter will yield data which can be evaluated in terms of loudness with reasonable engineering accuracy.

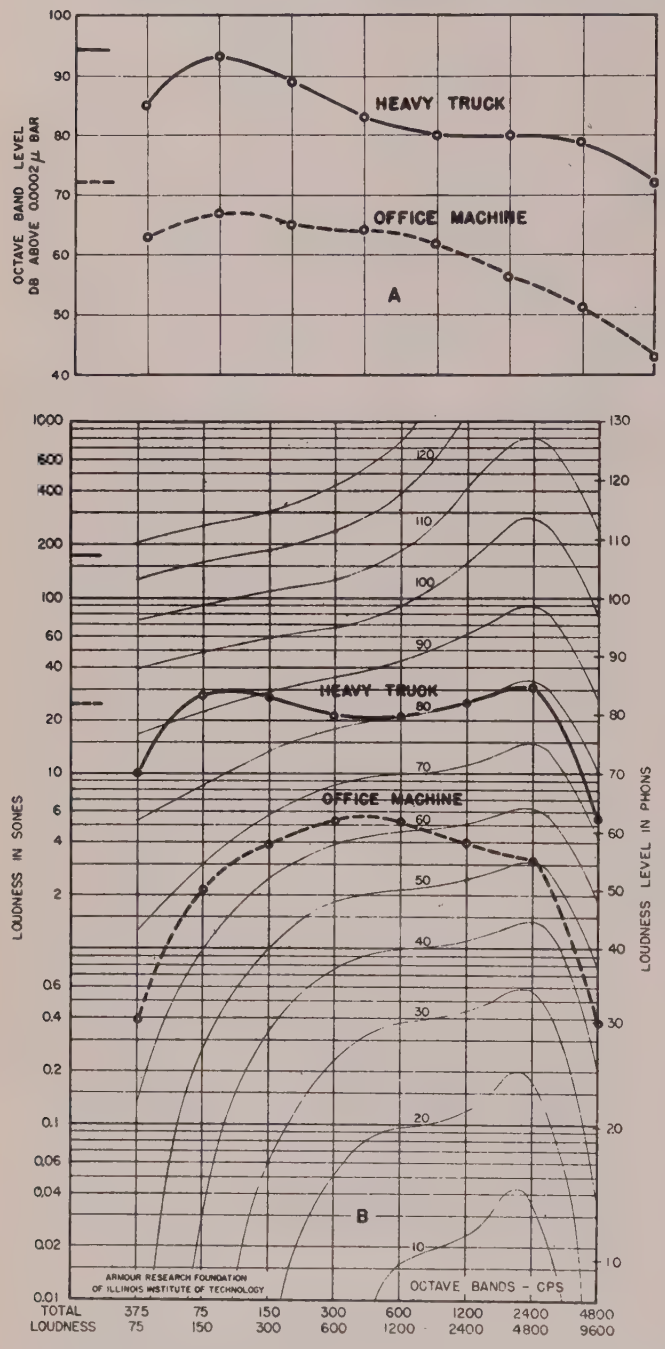

Fig. 1 – Octave band sound pressure levels for heavy truck and typical office machine. Sound pressure levels plotted in rectangular coordinates are shown in A (above), while the same levels plotted according to sound pressure contours are shown in B (below). By use of the latter diagram, loudness in each octave band can be read from scale at left and total loudness in sones can be determined by simple addition of individual octave band loudnesses

The frequency analyzer customarily used is referred to as the octave band analyzer, specifications for which have recently been promulgated in ASA Standard Z24.10. The electrical output of a standard sound level meter, which represents the total sound pressure of a complex noise, is presented to an octave band analyzer consisting of a series of contiguous filters covering essentially the audio frequency range. By means of a selector switch the sound pressure level is indicated successively on an output meter for each octave frequency band. This operation gives a rough picture of the frequency distribution of the noise energy such as shown in Fig. 1. While such data are useful in noise reduction problems and yield other useful information on the noise in question, it would be much more helpful if the data for each noise could be reduced to some single number which reasonably represents the total loudness. Methods of doing this have been described.[2-6] One method is based on the use of graph paper shown in Fig. 1. The octave band sound pressure levels are plotted according to the contours which represent sound pressure levels of equal loudness in octave bands. The loudness contribution of each octave band is then read from the scale on the left in units of sones. The total loudness of the noise can then be obtained by a simple addition of the loudness contributions by each band. For example, in the curve shown, the total loudness for a particular truck measured at a distance of 50 ft is approximately 175 sones, while the total loudness of a typical office machine is about 25 sones. The sone is an arithmetic unit rather than a logarithmic unit, like the decibel, so that a sound of 175 sones is about seven times as loud as one of 25 sones. This type of representation seems to be advantageous in comparing test measurements with results of psychological tests and is an easier concept than a logarithmic unit for the layman to understand.

If we assume that the technique described briefly above yields loudness values which agree with judgments by listeners, it is still a fair question

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[2] See *Journal of the Acoustical Society of America*, Vol. 23, May, 1951, pp. 261-269: "Calculation and Measurement of Loudness of Sounds," by L. L. Beranek, J. L. Marshall, and A. L. Cudworth.

[3] See p. 524 of "Acoustic Measurements," by L. L. Beranek. Pub. by John Wiley and Sons, Inc., New York, 1949.

[4] See *Journal of the Acoustical Society of America*, Vol. 24, January, 1952, pp. 80-82: "Loudness Chart for Octave-Band Data on Complex Sounds," by F. Mintz and F. G. Tyzzer.

[5] See pp. 72-73 of "Handbook of Noise Measurement," by L. L. Beranek and A. P. G. Peterson. Pub. by General Radio Co., Cambridge, Mass., 1953.

[6] See Proceedings, Second National Noise Abatement Symposium, Vol. 2, October, 1951, pp. 64-74: "Instrumentation and Techniques for Measurement and Evaluation of Industrial Noise," by D. B. Callaway.

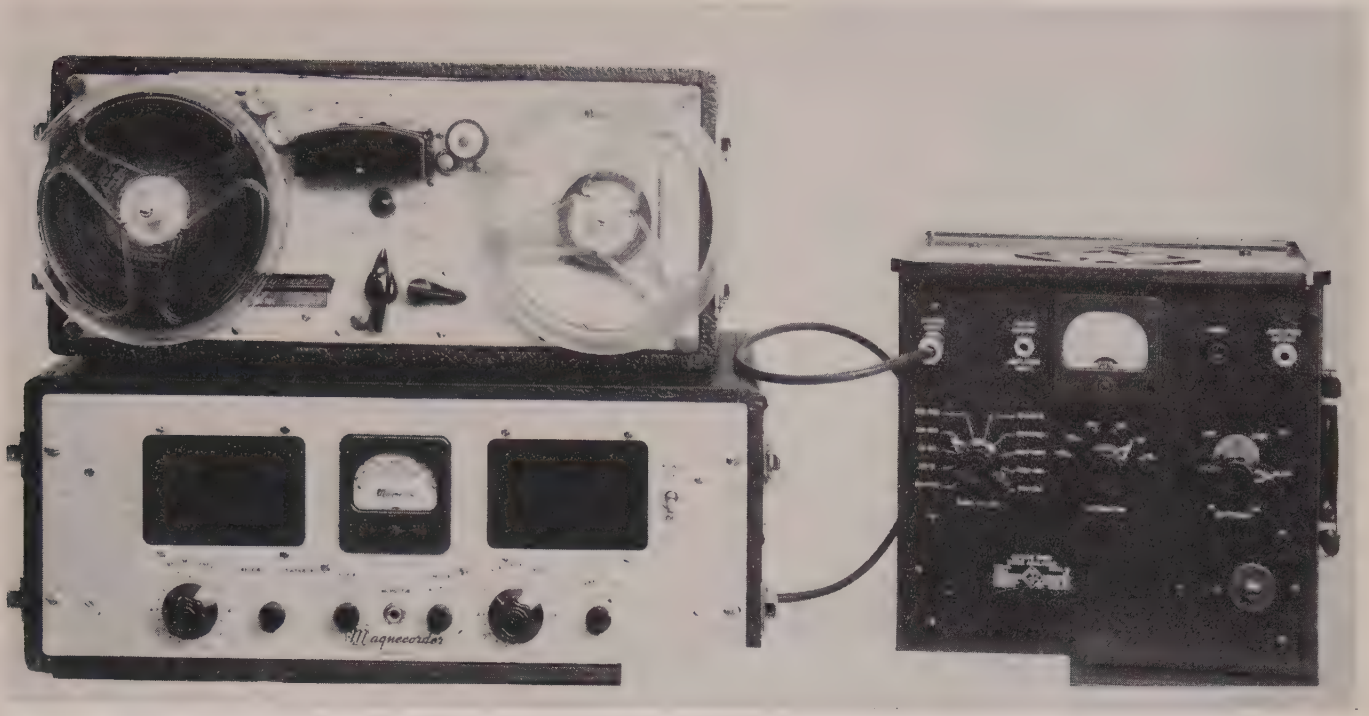

Fig. 2 – Photographic view of magnetic tape recorder connected to octave band frequency analyzer. By playing tape recordings of truck noise repeatedly into octave band analyzer, sound pressure level in each octave band can be obtained

to ask how a series of eight octave band measurements can be obtained and the loudness calculation performed in the short period of time required for a truck to pass a stationary observation point. The solution to this problem lies in the use of a magnetic tape recorder to obtain a record of the total noise of the truck. This recording can then at leisure be fed into the octave band analyzer, thus obtaining the required data for the loudness calculation. Such a tape recorder and an octave analyzer are shown in Fig. 2. It is estimated that a trained technician could, if required, complete the entire operation for a particular truck within 5 or at the most 10 min.

It can be argued, however, that this technique is still too complicated for use by law-enforcement officials in determining whether any truck exceeds a proposed loudness standard. This point is granted. It is our opinion that the best solution to the truck exhaust noise problem does not lie in placing measurement equipment in the hands of law-enforcement officials, but rather by the certification of mufflers for various types of trucks which have been tested by a qualified agency and found to meet an approved noise standard. This would allow simple legislation requiring that trucks operating on the highway be equipped with approved mufflers in good condition and would reduce the task of an enforcement official to that of examination of muffler for certification and condition.

Because of inconclusive or negative results obtained in past experiments attempting to relate subjective appraisal of truck noise with noise measurement data, it was decided to undertake a limited series of listening tests to validate the measurement techniques described earlier. Approximately 100 recordings of passing trucks were made on U. S. Route 66 near Chicago. The site chosen was approximately a 5% grade on which over-the-road trucks were passing in high gear. An approximately random distribution of loudness was obtained by recording all truck-trailers passing in the desired direction, the noise of which was not obscured either by preceding or following trucks or by trucks passing in the opposite direction. The recordings were made over a 12-hr period from early afternoon to nearly midnight. The loudness of each truck was calculated in the laboratory by the methods described above. Loudness judgment tests were then made by a panel of observers, using the same recordings.

Precise determination of loudness values by means of psychoacoustic tests would involve elaborate instrumentation, large numbers of observers, and many repeated experiments under carefully controlled conditions. Such elaborate tests were considered unnecessary to show good correlation between calculated and judged loudness values. The tests performed were quite simple and no elaborate psychoacoustic controls were employed.

The listening tests were conducted in a fairly live conference room at Armour Research Foundation. The recorded truck sounds were reproduced over a good quality audio system. The average sound level at the listening positions in the room

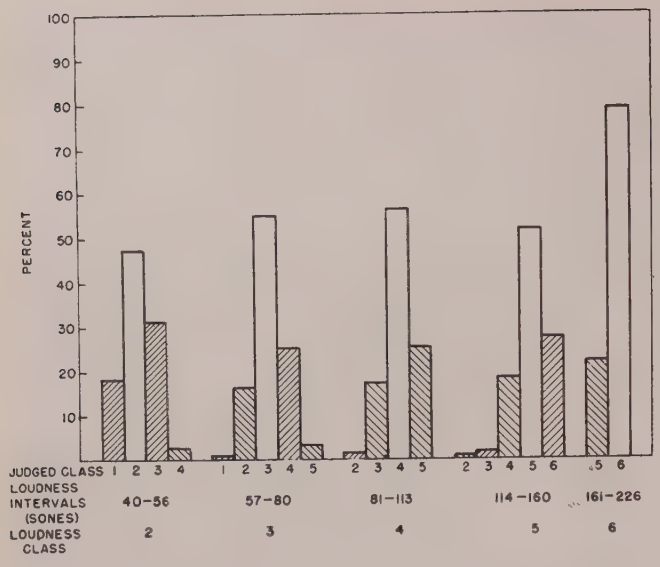

Fig. 3 – Distribution of truck loudness judgments in individual loudness classes; 50% or more of the loudness judgments are in proper class, while 98% or better are correct within one class

was adjusted to be equal to the original level at a distance of 50 ft from the traffic lane.

To enable the listeners to establish a loudness scale with which to evaluate the loudness of each truck noise, a separate recording was made representing trucks of the maximum and minimum loudnesses occurring in the test run. These standardizing signals were reproduced through the same speakers before each test and at intervals during each test to help the listeners keep the extremes of the loudness scale in mind. The minimum reference signal was equal in loudness to an average passenger car.

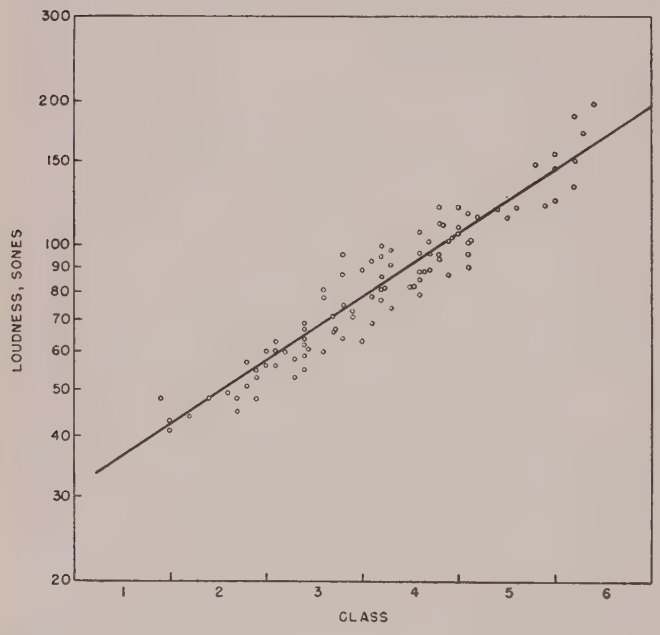

Fig. 4 – Correlation between calculated loudness in sones and average class judgments for 15 observers. Correlation coefficient 0.94

Two listening panels were used, each taking the test twice. The first consisted of ten people recruited from the acoustics section of the Physics Research Department. The second panel of five people consisted of representatives of the American Trucking Associations and members of the Physics Research Department of the Foundation.

At the beginning of each test, the subjects were asked to estimate and record the loudness of each truck on a scale of 1 to 6; in other words, to place each truck in one of six classes according to its loudness. They were informed that the reference sounds were in the upper part of class six and the lower part of class one. After these signals were presented, a group of recorded truck noises was reproduced in a practice period to allow the listeners to become familiar with the procedure and accustomed to the method of presentation. Then the reference sounds were repeated and the test truck noises presented immediately while the listeners recorded their judgment of the loudness of each successive truck. Short rest periods were permitted after each group of 25 trucks, and the reference sounds were repeated before proceeding with the following group.

Examination of the data showed that the differences among the observers were sufficiently small to permit averaging the results, thus reducing the number of pieces of data to be handled from 1500 to 100. The group loudness judgments are summarized in Fig. 3. This figure shows the percentage of judgments for all observers by classes which were exactly correct and within one class high or low of correct. The graph shows that for all loudness classes 50% or better of the judgments were exactly correct and that 98% or better were within one class of correct. Considering the inexperience of many of the observers, the number of trucks whose loudnesses were close to the divisions between classes, and the relatively crude nature of the physical and psychoacoustic controls, these results are considered excellent.

As pointed out earlier, the listeners were instructed to estimate the loudness of the recorded truck noise on a scale of 1 to 6. This scale was chosen for several reasons. Preliminary attempts to judge loudness in the field at the time the recordings were prepared were made using 10 arbitrary loudness divisions. It was apparent that these classes were too small for convenient use. In judgment tests, observers expressed uncertainty in classification and a study of the test results showed much overlapping between classes. Since it was anticipated that recordings of Pacific type trucks would provide noise samples higher than any obtained on this run, the 1-6 scale was chosen so that a total of 10 scaled divisions would not be exceeded for all trucks. Examination of the preliminary field loudness judgments showed that most of the observers tended to make their arbi-

trary scale divisions in equal percentage increments of loudness. It was decided, therefore, to divide the total range of truck noise into six bands which resulted in bands of approximately 40% loudness increase. This resulted in the following classifications for the calculated loudnesses:

Class 1: 28- 40 sones
Class 2: 41- 56 sones
Class 3: 57- 80 sones
Class 4: 81-113 sones
Class 5: 114-160 sones
Class 6: 161-226 sones

As pointed out earlier, it was decided on the basis of the small spread of the data to average the judgments of the 15 observers for the second test run. These averaged judgments were then correlated with the calculated loudness. The scatter diagram showing the relationship between these two variables is shown in Fig. 4. The vertical scale is calculated loudness in sones, while the horizontal scale indicates the average class judgment of 15 observers. Each point on the chart represents a particular truck. The scatter of these points indicates the departure from a perfect linear relationship between calculated loudness and loudness judgment. The correlation coefficient between these two variables, a term commonly used in statistical analysis, is a numerical measure of the degree to which two variables are linearly related. If all points fell on a straight line, the correlation coefficient would be 1.0, and the linear function would enable us to compute loudness exactly from octave band measurements. The linear relation which fits the average of all data points is represented graphically by the regression line. A correlation coefficient of 0.50 would indicate a uniform tendency of data points about the regression line but would be of little value in establishing a definite relation between subjective loudness judgment and calculated values. The correlation coefficient for the data shown in Fig. 4 was computed to be 0.94.

After determination of this rather high correlation, it was thought to be of interest to study the correlation between the loudness judgments and the overall sound pressure level measurements for the 97 trucks both on the flat network and the 40-db weighting network.

Fig. 5 shows the scatter diagram for the overall level on the flat network. General comparison of the scatter on this chart with that of Fig. 4 indicates the improvement offered by the loudness calculation method over the traditional measurement of sound pressure level and explains the reason for previously reported disagreement between measurements and judgments. For example, trucks on which the average loudness judgment ranged from class 3 to class 5, all had the same sound pressure level, or looking at the chart along a vertical section, trucks of the same average class had sound pressure levels which varied more than

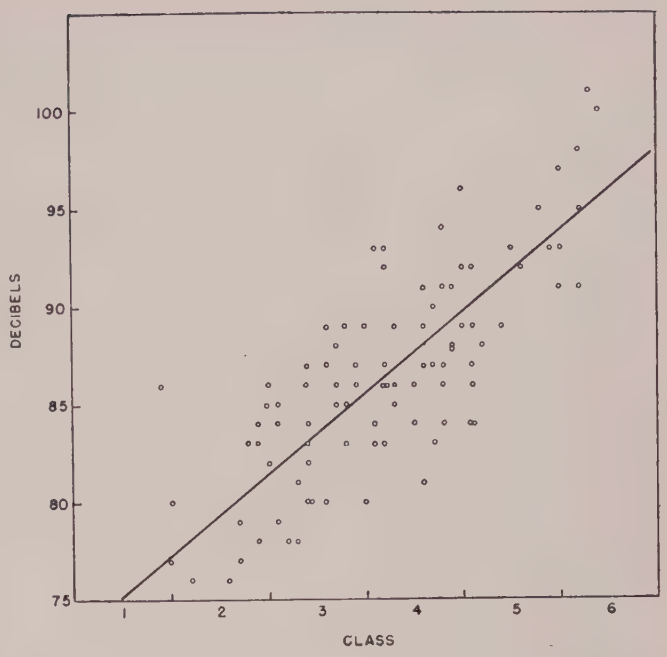

Fig. 5 – Correlation between sound pressure levels on flat network and average class judgments for 15 observers. Correlation coefficient 0.78

15 db. The correlation coefficient for these two variables was found to be 0.78, considerably lower than that for the calculated loudness values.

A similar scatter diagram relating the average loudness judgment to the sound pressure level measured on the 40-db weighting network is shown

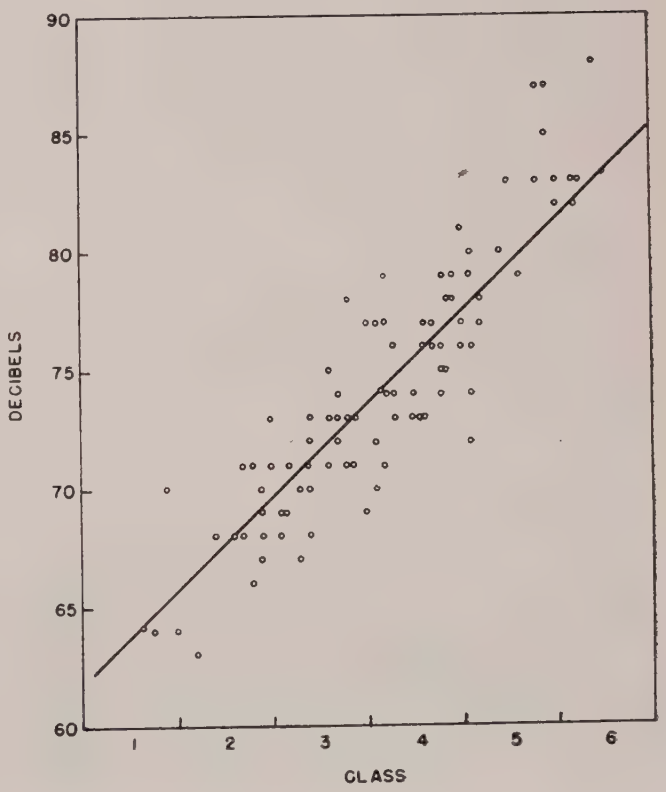

Fig. 6 – Correlation between sound pressure levels on 40-db network and average class judgments for 15 observers. Correlation coefficient 0.83

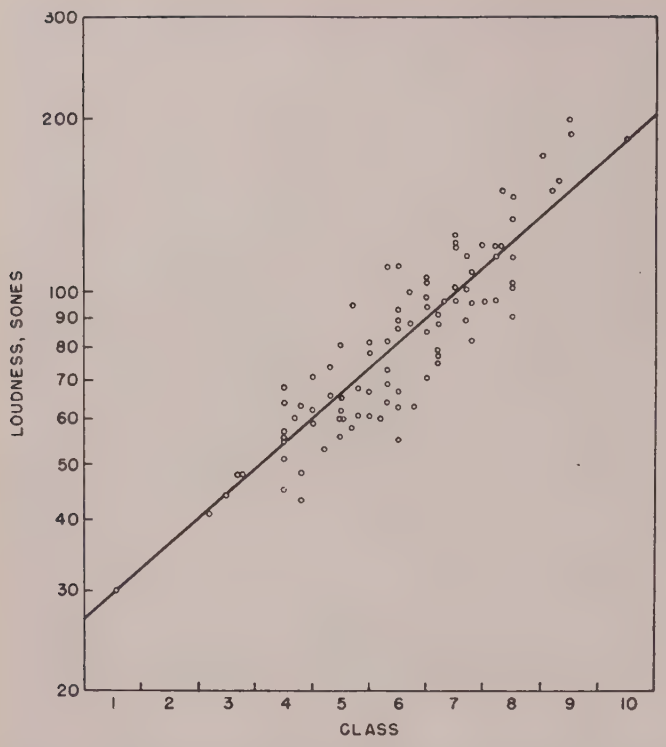

Fig. 7 – Correlation between calculated loudness and average roadside loudness judgment for four observers. Correlation coefficient 0.87

in Fig. 6. This correlation was attempted because even though the instructions for use of the sound level meter would recommend use of the flat network, it appeared that the 40-db network would de-emphasize the low frequencies in a manner similar to that provided by the equal sound pressure contours on the loudness calculation paper shown in an earlier figure. In this case, the correlation coefficient was 0.83, somewhat better than that for the overall on the flat network, as expected, but considerably lower than that for the calculated loudness values.

There is one more scatter diagram to examine, and this is shown in Fig. 7. This shows the relationship between calculated loudness and average class judgment for four observers made in the field at the time of preparation of the recordings.

Fig. 8 – Photographic view of road test location on U. S. Route 66 near Chicago

In this case, the correlation coefficient was 0.87. This indicates that roadside judgments are not as good as loudness judgments made under controlled conditions but are better than the use of the sound level meter readings for determining loudness.

Early in the course of this project it was agreed that the scope of a survey of truck exhaust noise conditions would necessarily be limited to specified test conditions and to a few measurement locations. On the basis of discussions with trucking company representatives it was concluded that a reasonable sampling of over-the-road trucks could be obtained from measurements over a 24-hr period at one or two typical locations in the Middle West and on the West Coast. This conclusion was reached as a result of the general opinion that the West Coast truck noise problem is more severe, while that in the Middle West is quite similar to that in all other parts of the country. It was agreed that the operating conditions which should be measured should be of over-the-road carriers pulling up a medium grade in cruising gear. Test locations were to be chosen at which traffic was heavy and a random sample was to be obtained by measuring approximately 100 trucks selected at random over all times of the day. The main location chosen for the Middle West test was on U. S. Route 66, approximately 20 miles southwest of Chicago. Trucks traveling west were measured while climbing a 5% grade. By measuring all trucks which passed without traffic preceding, following, or passing in the opposite direction, over a period from noon until midnight, a fairly random sample was obtained. A small fraction of the Middle West measurements were made at a second location on U. S. Route 83 on a 4% grade which was somewhat less heavily traveled.

The West Coast location chosen was near Portland, Oregon, on U. S. Route 99. Most of the recordings were made at a location about 5 miles from Portland, with trucks southbound, climbing approximately a 5% grade. These recordings were made between about 2 and 10 p. m. A smaller fraction of the recordings was made on the same highway on a similar grade inbound toward Portland between the hours of 4 and 8 a. m. Test locations were chosen which were reasonably free from large buildings, high reflective banks, and interfering noise sources.

The instrumentation used for the field tests consisted of a dynamic microphone and a high-quality magnetic tape recorder driven by a dynamotor powered by a 12-v d-c supply. A sine wave calibrating signal representing a sound pressure level of 100 db was recorded at the beginning of each roll of tape. The measuring microphone was located 5 ft above the ground at a distance of about 50 ft from the traffic lane and was covered by a silk screen to prevent interference due to wind. A photograph of the field test location used for the Chicago tests is shown in Fig. 8.

These recordings were analyzed at a later time

in the laboratory by determining the sound pressure levels in octave bands. From these data the loudnesses of the individual trucks were computed by the method described earlier.

The distribution of loudness for Middle West and West Coast trucks divided into loudness classes is shown in Fig. 9. The vertical bars represent the percentage of trucks in each loudness class where loudness was calculated by the methods described earlier. The total number of trucks considered in the Middle West case was 97, while for the West Coast the total number was 94. Examination of this figure shows that for the Middle West the loudness distribution centers near class 4 with 35% of the trucks measured in this class. The distribution is slightly asymmetric with 29% of the trucks measured in class 3, and 16% in class 5. For the West Coast trucks, the loudness distribution centers near class 5, with 40% of the trucks measured in this category. Of the West Coast trucks, 32% were in class 4, and 21% were in class 6. About five trucks were in class 7, but of these, three were close to the border between classes 6 and 7. Only one truck, however, was in class 2 and one in class 3.

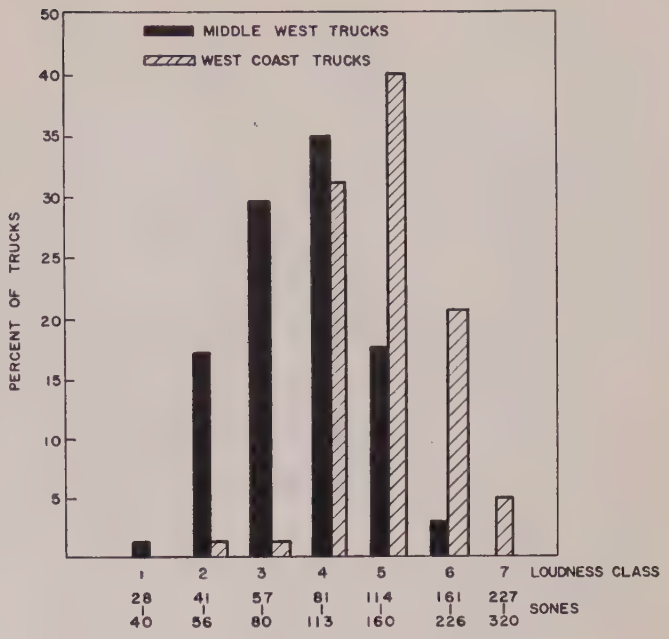

Fig. 9 – Loudness distribution for Middle West and West Coast trucks

Comparison of the overall distribution indicates that there are only a few West Coast trucks which are greatly louder than the loudest trucks found in the Middle West, but that the average loudness of West Coast trucks is more than one class higher than for the Middle West.

The differences between the loudness distribution curves suggested an examination of the typical octave band levels for trucks of the two areas. The median sound pressure level for each octave band and the upper and lower quartile limits for each octave band were therefore determined for the Middle West and West Coast cases. The range between upper and lower quartiles includes 50% of the trucks measured. These octave band levels are plotted in Fig. 10 as curves 1 and 2. Comparison of the median values indicates a difference of only 3 to 5 db in the three low-frequency and the two high-frequency bands with differences of the order of 10 db in the octave bands between 300 and 2400 cps. In all octave bands, however, the median for the West Coast trucks is higher than that for trucks measured in the Middle West. Also plotted in Fig. 10 are the median and quartile limits for 17 passenger cars traveling on a level highway at speeds between 50 and 60 mph. These curves will be discussed in a later section.

After obtaining the median and quartile octave band levels, the median and quartile loudnesses for Middle West and West Coast trucks were determined by means of the loudness curves shown in Fig. 11. This figure indicates that the median loudness for West Coast truck noise is about 130 sones, while that for the Middle West is about 85 sones, approximately one class lower. The median for 17 passenger cars, however, is only 24 sones.

The purpose of the tests described below was to investigate the possibility of predicting from noise measurements either on an engine test stand or a chassis dynamometer the levels which would result from truck exhaust noise at a specified distance from the truck on the highway. These tests were conducted in Portland, where facilities of Consolidated Freightways, Inc., were available.

All of the controlled tests were made with a Buda type DAS 280-hp 6-cyl diesel engine operating at 1800 rpm under full load. This condition was chosen because with diesel operation drivers make every effort to maintain optimum rpm under all road conditions and also because it appears

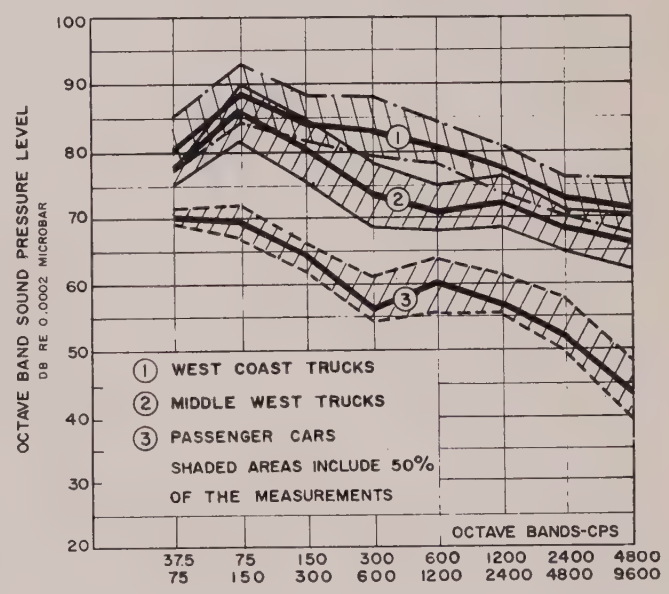

Fig. 10 – Median and 50 percentile values for West Coast trucks, Middle West trucks, and passenger cars

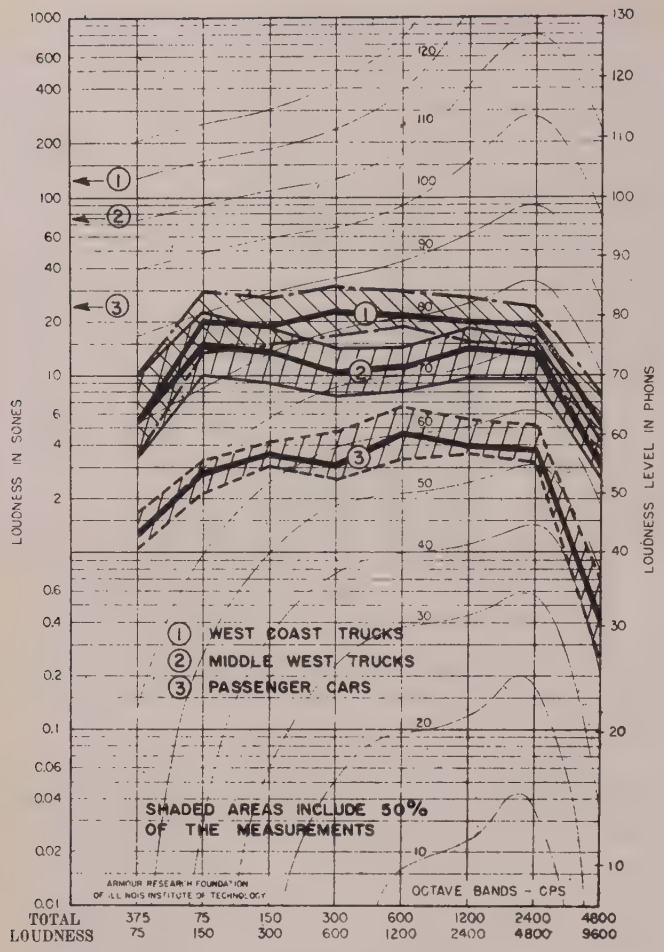
LOUDNESS IN SONES
LOUDNESS LEVEL IN PHONS
① WEST COAST TRUCKS
② MIDDLE WEST TRUCKS
③ PASSENGER CARS
SHADED AREAS INCLUDE 50% OF THE MEASUREMENTS
ARMOUR RESEARCH FOUNDATION OF ILLINOIS INSTITUTE OF TECHNOLOGY
OCTAVE BANDS - CPS
TOTAL LOUDNESS
375 75 150 300 600 1200 2400 4800
75 150 300 600 1200 2400 4800 9600

Fig. 11 – Loudness curves for West Coast trucks, Middle West trucks, and passenger cars

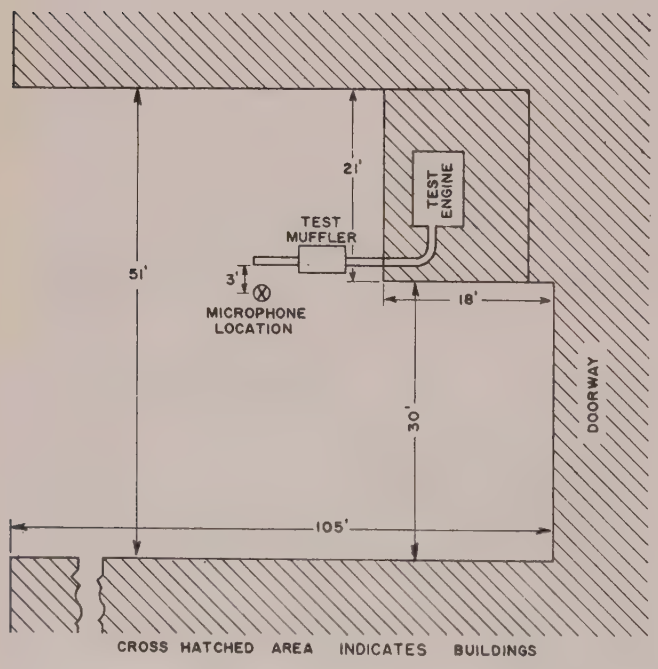

Fig. 12 – Plan view of areas in which engine dynamometer muffler tests were conducted

that the exhaust noise is practically independent of the road speed as long as the engine is operating at rated rpm under full horsepower. The chassis dynamometer tests were made in an enclosed building barely large enough to accommodate a Consolidated Freightliner van. Since the test room was not acoustically treated, it was necessary to make chassis dynamometer noise measurements with a microphone location only 1 ft horizontally displaced from the end of the tailpipe. This location was chosen in order to assure that the noise measured was due to engine exhaust and was not radiated from other parts of the truck or exhaust system, or reflected from the room surfaces. For the chassis dynamometer tests, as well as those made on the engine dynamometer and on the road, three test conditions were measured: (1) with no muffler in the system, (2) with a muffler providing medium noise reduction, and (3) with a muffler believed by Consolidated personnel to be superior in noise-reducing effectiveness.

Engine dynamometer tests were made using another Buda diesel engine of the same type and horsepower, mounted on an engine test stand inside a masonry structure. The exhaust pipe was led through a hole in the enclosure to a fairly open area out-of-doors. The pipe was brought through the wall at a level approximately 1 ft above the ground. Each test muffler was installed at an average distance of 5 ft from the building wall. The general configuration of the area in which the tests were conducted, shown in Fig. 12, is quite far from ideal acoustically. The shaded areas shown are surrounding buildings, and the ground was paved with macadam. Because of the differences in the design of the various mufflers tested, it was not always possible to maintain a standard configuration for the portion of the exhaust system outside the engine test cell. Furthermore, it is believed that reflections from the macadam and from the various building walls may have influenced the test results. The tests were considered worth while, however, because of the fact that most engine or muffler manufacturers would be more likely to have such a facility available than a chassis dynamometer and because it was felt that the data obtained from the tests would be of value in designing a test facility with adequate acoustical properties. The road tests were made in the same manner as described earlier for the survey, with the test truck passing the microphone location at a speed of approximately 35 mph.

The octave band sound pressure levels at a distance of 1 ft in a horizontal direction from the end of the vertical tailpipe on the chassis dynamometer, 3 ft in a horizontal direction from the discharge end of the tailpipe on the engine test stand, and approximately 50 ft from the end of the exhaust stack as the truck passed on the highway, were measured for three conditions: (1) without muffler, (2) with a muffler of intermediate quality referred to throughout this paper as "muffler A,"

and (3) with a more effective muffler referred to throughout as "muffler B." The results of these tests with no muffler are shown graphically in Fig. 13 by the solid curves. Octave band levels between 115 and 123 db exist at a 1-ft distance from the tailpipe without muffler. The reduction due to distance results in levels at 3 ft of approximately 110 ±3 db in all octave bands, while the levels measured at 50 ft are between 80 and 90 db with the lower levels in the higher frequency bands. The octave band levels which would be predicted from calculation of the reduction due to distance for distances of 3 ft and 50 ft are shown by dashed curves in Fig. 13. The agreement between the actual measurements and the calculated levels is considered to be excellent for the test conditions which existed. For example, the loudness of this truck at a distance of 50 ft calculated from the measurements at 50 ft, was 217 sones, while the loudness obtained from the calculated levels is 213 sones. This indicates the accuracy which can be expected from the method of calculation and also shows that the noise measured on the highway was entirely due to that from the stack and not due to other noise sources on the truck.

Fig. 14 shows the same information for the truck and test engine equipped with muffler A. In this case, the calculated levels at the 3-ft distance are in good agreement with those measured on the engine dynamometer stand, but the levels calculated for the 50-ft distance are 3 to 10 db low, with the largest errors in the low-frequency octave bands. The loudness at 50 ft, based on the actual measurements, was 160 sones, while that determined from the calculated levels for 50 ft was only 93 sones. The only conclusion which can be drawn from this discrepancy is that there are other sources of noise on the truck, possibly including sound coming directly through the muffler or exhaust pipe sidewalls which contribute to the level at a distance and which, therefore, place a lower limit on the noise reduction which can be achieved by the muffler.

This fact is further demonstrated in the test results for muffler B, which was even more effective than muffler A. In this case, the levels at 1 ft on the same engine are slightly greater than 100 db in the middle-frequency octave bands decreasing to less than 90 db for the high-frequency bands, indicating appreciable noise reduction by this muffler. The measured levels at 3 ft and at 50 ft are shown in Fig. 15 and can be compared with the calculated levels for those distances indicated by the dashed lines of the curve. In this case, good agreement between measured and calculated levels is obtained for the 3-ft distance, except for the two highest frequency octave bands. These discrepancies are probably due to reflections from the macadam driveway or building surfaces which existed near the measurement location for the engine dynamometer tests. For the 50-ft distance, the calculated levels are 5 to 10 db lower

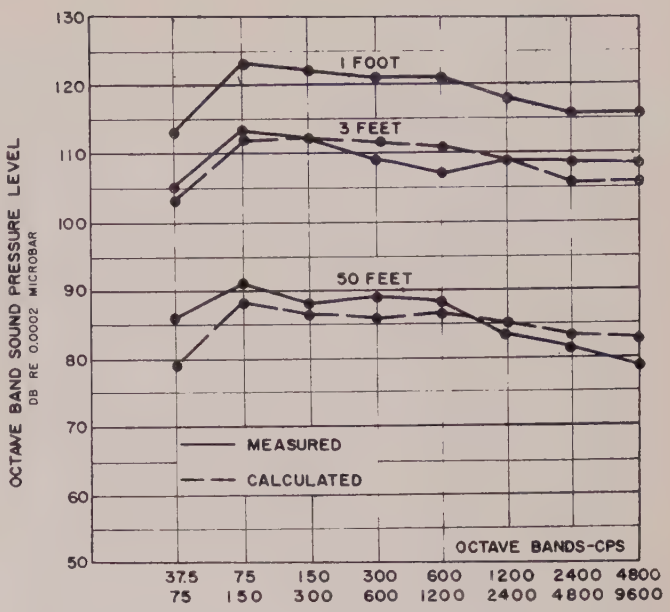

Fig. 13 – Octave band levels for 250-hp diesel, 1800 rpm, no muffler

than those measured, indicating again that other noise sources are contributing to the total noise levels on the highway. In this case, the loudness measured at 50 ft was 66 sones, while that calculated on the basis of measurements at 1 ft was only 40 sones.

As stated earlier, the reason for the survey of truck noise conditions was to form an intelligent basis for standards to be used by the trucking industry which would eliminate or reduce to a tolerable number the complaints arising from noise of over-the-road trucks. Such standards must be prepared by groups directly concerned with the problem, but some comments regarding factors

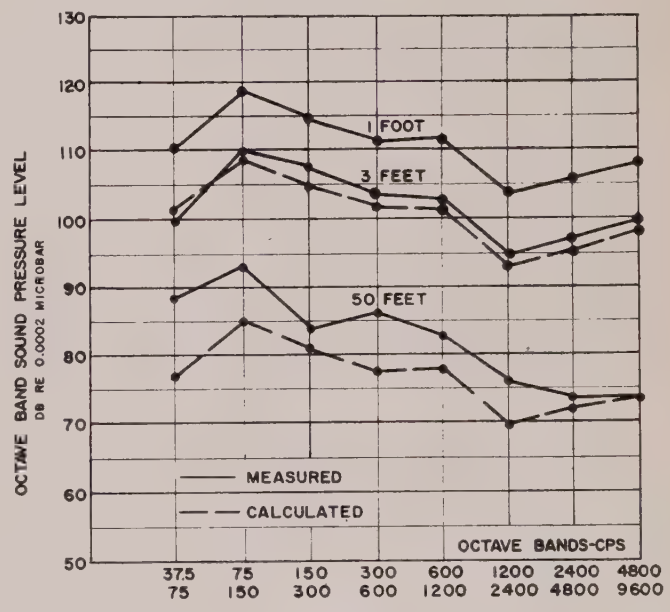

Fig. 14 – Octave band levels for 280-hp Buda diesel, 1800 rpm, muffler "A"

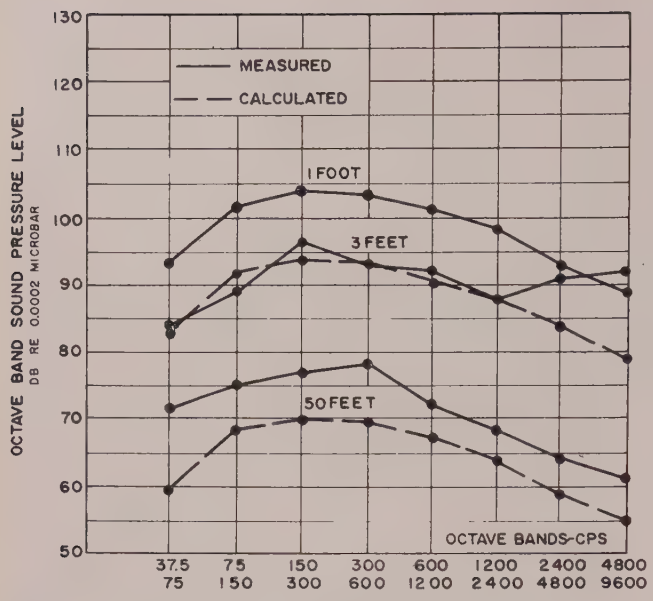

Fig. 15 – Octave band levels for 280-hp Buda diesel, 1800 rpm, muffler "B"

involved in their determination can be made.

In order to establish limits for the loudness of truck exhaust noise which will eliminate complaints, it would be helpful to study the loudness of similar noises which are generally accepted without complaint by residents along the highway. Shown in Fig. 10, which was discussed earlier, are the median octave band levels for the West Coast trucks, the Middle West trucks, and 17 passenger automobiles driving at 50 to 60 mph on Edens Expressway near Chicago. The median loudness for these passenger cars was 24 sones, about 1/6 the loudness for the median West Coast trucks and nearly 1/3 the loudness for the Middle West trucks. For several reasons, however, it is not considered realistic to prepare a standard limiting the noise of trucks to that for average passenger cars: (1) the data on the 17 passenger cars shown in Fig. 10 were for cars cruising on level highway and are probably somewhat lower than for automobiles pulling a grade similar to those on which the truck tests were made; (2) the curves of Fig. 10 include only 50% of the measured passenger cars, so that 50% will be louder than the 24-sone median figure; (3) it is believed that the general public will accept as a satisfactory noise level from a large truck something considerably higher than the median values for a typical modern passenger car.

On the basis of the truck noise survey, the per cent of the vehicles which would require better silencing for various limiting values of truck noise can be predicted. Table 1 shows the percentage of Middle West and West Coast trucks which would be louder than the standard if the standard were set up as the upper limit of loudness classes 3, 4, and 5. The table shows, for example, that if the upper limit for the loudness of truck noise at 50 ft were set at 80 sones, the top limit of class 3, 50% of the Middle West trucks and 98% of those measured on the West Coast would require additional silencing. This may seem a drastic step which would result in a great economic burden for the trucking industry. On closer examination, however, such a limit does not appear unreasonable. In the first place, this would allow trucks to be approximately three times as loud as the median loudness of high-speed passenger automobiles, a ratio which, in our opinion, is near the upper limit which people will tolerate without many complaints. Furthermore, the controlled tests on the 280-hp Buda diesel indicate that large trucks can be silenced to such levels with commercially available, reasonably inexpensive, serviceable mufflers.

It may be thought that since other noise sources on the truck are only 10 to 15 db below the exhaust noise without a muffler, the use of a muffler providing 20 db or more noise reduction would be unnecessary. It is believed, however, that maximum exhaust silencing is desirable in order to reduce further exhaust noise when the vehicle is operated at low speeds in congested city areas, since under these conditions other noise sources on the truck are probably less important.

As pointed out at the beginning of this paper, one of the objectives of the work discussed in this report was to demonstrate the feasibility of predicting loudness of truck exhaust noise on the highway from measurements made under controlled laboratory conditions. The reason underlying this objective is that measurement and evaluation of truck exhaust noise on the highway is a fairly complex task which could not easily be performed by law-enforcement personnel. If reasonable noise standards can be set and adequate laboratory measurements on mufflers can be made, some authorized organization could then evaluate various types of mufflers for use with different types of trucks, certifying as approved those which will result in meeting the established standard. The precedent for this method of obtaining satisfactory performance has been set by certification of sealed-beam lights, hitches, couplers, fire extinguishers, and other accessories.

The measurements described in this report indicate that the loudness of truck exhaust noise can be calculated with good accuracy when exhaust noise is sufficiently loud to override the noise due

Table 1 – Effect of Various Noise Level Standards on Silencing Requirements

Limiting Loudness, sones	Highest Loudness Class Allowed	Per Cent of Trucks Excluded	
		Middle West	West Coast
160	5	3	26
113	4	18	75
80	3	50	98

to other sources on the truck. When a good muffler is installed, the calculated loudnesses at 50 ft may be considerably lower than those which actually result if other components on the truck produce appreciable noise. This fact must, of course, be taken into account when writing standards, but it is believed that few trucks will result in objectionably loud noise levels if the exhaust noise contribution is essentially eliminated.

The tests described earlier indicate that additional research is needed to specify the manner in which test-stand measurements of exhaust noise should be made. To the best of our knowledge, no test stand has been built primarily with noise measurement considerations in mind, and considerable planning and engineering design are required before an adequate installation of this type can be constructed. For example, it is believed that any such facility should produce repeatable results to within ± 2 db in all octave bands and should allow evaluation of noise escaping through the muffler sidewalls as well as that emerging from the tailpipe. Such installations must be adequately instrumented and should incorporate sufficient flexibility to permit reasonably easy tests.

There are undoubtedly other problems and considerations involved in design of such a test facility, and it is believed that after such a facility is built, some additional measurements under controlled road conditions should be made to validate results obtained on the test installation.

On the basis of the work which has been discussed, the following conclusions can be drawn:

1. Measurements can be made with commercially available instrumentation which yield results in good agreement with listening tests of the loudness of truck exhaust noise.

2. The techniques recommended are sufficiently simple to be used by trained technicians but do not lend themselves to immediate evaluation of the loudness of passing highway trucks.

3. On the basis of the survey of Middle West and West Coast truck noise, it is concluded that West Coast trucks average about 55% louder than those of the Middle West, which is equivalent to somewhat more than one loudness class. Of the Middle West trucks, 35% were found to be between 81 and 113 sones, and 40% of the West Coast trucks were between 114 and 160 sones at a distance of 50 ft.

4. When exhaust noise is the main contributor to the noise of a truck heard on the highway, the noise levels which will exist can be calculated with good accuracy from test-stand measurements. With good mufflers, however, other running noises result in noise measurements on the highway which are generally higher than would be calculated from test-stand measurements.

5. On the basis of measurements on a large diesel truck equipped with a muffler of present commercial design, it may be practical to limit exhaust noise at 50 ft to a value in the neighborhood of 80 sones. This loudness is believed to be higher than that which will result from other noise sources on most trucks. A loudness limit of 80 sones would require improved exhaust silencing for about 50% of the Middle West trucks and for nearly all large trucks operating on the West Coast.

Acknowledgment

The work discussed was conducted under the supervision of H. C. Hardy, whose helpful suggestions are appreciated. F. G. Tyzzer contributed materially to the analysis of the test results. H. H. Hall planned and conducted much of the experimental work. The assistance of Lewis Kibbee of the American Trucking Associations and of E. B. Ogden, Consolidated Freightways, Inc., and his staff, is gratefully acknowledged.

DISCUSSION

Sound Quality versus Sound Loudness

– L. H. Billey
Donaldson Co., Inc.

PERHAPS other interested people felt as we do, that the 3-band meter may have been in the right direction but was far from the answer to the sound measurement problem.

It is unfortunate that the American Trucking Associations could not accomplish more and sooner because the highway noise problem has become an important political item in the majority of rural communities in these United States. Rural (and urban) legislative aspirants are all promising "to do something" about truck noises. Many have risen to office on this issue.

We have relegated "loudness" to a second position of importance. We believe sound "quality" is first in order. In our work we first try to develop sound quality out of this noise and then reduce loudness. We believe that one can tolerate a loud sound of pleasant quality more readily and longer than an unpleasant sound of much lower "loudness level." We can hear the local city band practicing in the summer evening fully as far as we can hear the truck tanker, and it doesn't bother us as much either. We have

developed a sound quality contour curve (or curves, since these vary somewhat for widely varying noise levels). This curve is based on three factors:

1. The jury ratings of mufflers from tests conducted by the MTA of Southern California at Los Angeles.
2. The Fletcher-Munson equal loudness contours.
3. The mathematics of sound masking, which is represented in the square wave formulas.

We have our own complete, accurate, first-hand data, including facsimile tape recordings of all tests from the first to the fourth-and-last run by MTA of Southern California. We analyzed and used the jury ratings of the last tests for our psychological factors.

This sound quality curve is imprinted on graph paper. From the complete sound spectrum analysis (we do not feel that the octave-band meter qualifies here because we may and often do find a significant loudness peak and valley too close to one another in a given octave) we plot all the significant peaks and valleys. The fundamental or firing frequency is our base and all other frequencies are plotted their relative number of decibles above or below this base value. This quality curve requires that the second and third multiple of the firing frequency must fall into a definite range below the fundamental. It also requires that no band or ranges of frequencies shall be over-attenuated. Thus, if the sound should fall at all points on the curve it becomes a random noise and cannot be distinguished from any other random noise. It sounds more like a strong, steady wind blowing through the trees or a water-fall at a suitable distance. A true random noise is not too harsh to listen to.

The next and easy step then is to reduce the overall sound level. All we need is additional space or volume to do it in. We recognize that truck manufacturers place weight and space at a premium. But no one yet has found a substitute for volume in doing a good job of silencing—and that's what the public is demanding!

Rather than a tape recorder and an octave-band analyzer for rapid sound quality evaluation we would propose the tape recorder and a cathode-ray oscilloscope. The wave pattern of the sound, with but a little training, will reveal instantly to the eye those factors which distinguish a harsh noise from a pleasant one. As we approach a smooth sine wave form we approach a pleasant noise.

We agree heartily to a national central muffler testing facility. We would have only chaos if the several states set up independent and divergent testing schemes. The central, cooperative, nationally recognized testing laboratory is surely a better answer. Many automotive devices – headlamps and their settings are an example – must be of an approved type and adjusted by an approved adjuster. Muffler approval need not be this complicated.

We record all our tests but we have had little success with the recording of a moving vehicle from a stationary point. We need about 5 sec of uniform intensity sound for our analysis and at 40 mph, for instance, the truck would travel about 100 yd. That upsets our system of analysis. We do, however, record successfully with portable equipment in moving trucks, cars, or trains.

I sometimes think the shopworn excuse, "there is no instrument that measures noise like the human ear," had better be forgotten. The public knows when a muffler is too noisy and they don't use an instrument. That is our criteria – why cry for an instrument? If you can hear an annoying noise, it's too loud – go to work on it. If we don't, the legislators will. Instrumentation, size of muffler, problems of installation are no longer the main issue.

We have done a lot of listener or auditor evaluation but do not follow the system discussed. First we must distinguish between loudness value and annoyance value. Loudness alone is not a critical factor. It is the annoyance or nuisance factor (we call it A. Q. the annoyance quotient) that we need to evaluate. We make the so-called blindfold test recording tapes. We go directly from one noise to another without pause, space, or reference noise. The hearing mechanism has inertia or hysteresis and the best way to avoid its influence in hearing is to pass from one sound right to the next. The ear recognizes the change better at this instant than at any other time during the audition. We find in these tests that the auditors cannot judge loudness readily. What they hear and recognize is the annoyance comparison or A. Q. (annoyance quotient – when it is given a numerical value).

It is interesting to know that truck noise varies in different sections of the country. It provides us with information on where to commence improvements. The most stringent regulations are appearing east of the Rockies, however.

Exhaust noise in trucks and scavenger blower noises in many buses are by far the loudest and most annoying. These can be masked by good muffler engineering into an overall sound pattern in a manner that makes them indiscernible.

I cannot visualize any system of noise evaluation that a highway patrol officer could use and bring into a court of justice as evidence that someone is violating the noise statutes. He must pursue the methods now used in cases of suspicion of poor brakes or drunken driving and that is to arrest the driver and check him or his equipment. This is a natural and convenient way to check exhaust equipment too. All we need is an approved list of matching equipment truck, muffler, and exhaust system.

This system should provide for improvement as the art progresses. If we set specific sound levels as our goals two things result, first, we haven't evaluated the noise from its annoyance angle and second we set up a target which becomes the optimum goal of all muffler makers and truck builders. Progress stops there. Let's set up a testing laboratory, let's rate these products as Class A, B, C, and so forth, to give a user a choice of something passing or something even better.

Author's Closure To Discussion

A STUDY of Mr. Billey's comments indicates that we are in agreement on most of the conclusions stated in my paper. Our major difference appears to be with regard to the importance and validity of rating truck exhaust noise in terms of "loudness." We believe Mr. Billey has a point, but that it is of relatively minor importance. In the field of psychological acoustics, considerable experimental work has been done on the determination of loudness on various types of noise. Tentative loudness scales have been proposed, but further work is needed before standards can be written. The situation, with regard to annoyance scales, is much less satisfactory. Indeed, there is much doubt among many authorities as to whether satisfactory annoyance scales can ever be formulated. It is hoped that Mr. Billey will publish a detailed procedure for the determination of annoyance factors by physical measurements and graphical methods, because this is the subject of considerable controversy among acoustical experts.

At the present time, the most we can say is that there appears to be a direct correlation between loudness and annoyance and that for sounds of a given type such as those of truck exhausts, any reduction in loudness is accompanied by a reduction in annoyance. Because of the simplicity of the technique for determining loudness from octave band measurements, and the fairly wide acceptance this technique has already received, it seems unwise at this point to further complicate the situation by an additional system of numerology. Regardless of the measurement system which is ultimately accepted, we are convinced that truck noise can be measured with results that will make possible adequate rating scales for truck mufflers.

Airplane Configurations for High-Speed Flight

Clarence L. Johnson, Lockheed Aircraft Corp.

This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 2, 1953.

WITH the various recent improvements in aircraft powerplants, particularly the afterburning turbojet and the rocket motor, it is now possible to build aircraft which can reach supersonic speeds in level flight. Along with the powerplant developments has come the vast aerodynamic improvement of our airframes for high-speed flight.

This paper will discuss a few factors relating to the powerplant problem for high-speed aircraft, but will concentrate mainly on the problem of wing planforms and overall aircraft configurations for the Mach number range of 0 to 2.

Recently, there has been an extensive amount of publicity favoring delta-wing aircraft, both in this country and in England. With very little flight experience and, in some cases, very limited wind-tunnel data, many companies (including Lockheed) started drawing and building delta-wing aircraft. Trade journals seized on the trend, and most aircraft advertising started to feature unstable, tailless deltas. Styles are bad things in engineering. If this discussion does nothing more than promote further critical evaluation of wing and tail planforms, its purpose is served.

THIS paper discusses problems encountered in the Mach number range of 0 to 2 with aircraft having different wing planforms. The delta-wing type is compared to the unswept or straight thin wing.

It is concluded that, with the solution to the structural problem having been obtained for the latter wing, the unswept type is superior to the delta type when strictly comparative missions and operating field lengths are considered.

The Author

CLARENCE L. JOHNSON is chief engineer for Lockheed Aircraft Corp. He received a master's degree in aeronautical engineering from the University of Michigan. In 1940 Mr. Johnson received the SAE Wright Brothers Medal for his paper discussing work on control problems of four-engine airplanes. Prior to joining Lockheed in 1933, he was associated with Buick and Studebaker.

Some Basic Thrust Considerations

The afterburner attached to the turbojet engines has been viewed by many as a "contraption of the devil," due to its noise, complexity, and tremendous fuel appetite. In spite of these factors, the afterburner is here to stay, and today it is the most practical means for obtaining supersonic level flight speeds. We are often asked, "Why not just put a bigger, plain jet engine into your airplanes and forget the afterburner?" Fig. 1 shows why this is impractical.

If we consider two engines having the same static thrust, one equipped with an afterburner and one without an afterburner, the thrust output will vary with speed, as shown by the chart. At a Mach number of 2, the afterburning engine is developing 320% of its static thrust, while the nonafterburning engine is developing only 180%. This difference of 140% naturally gives marked per-

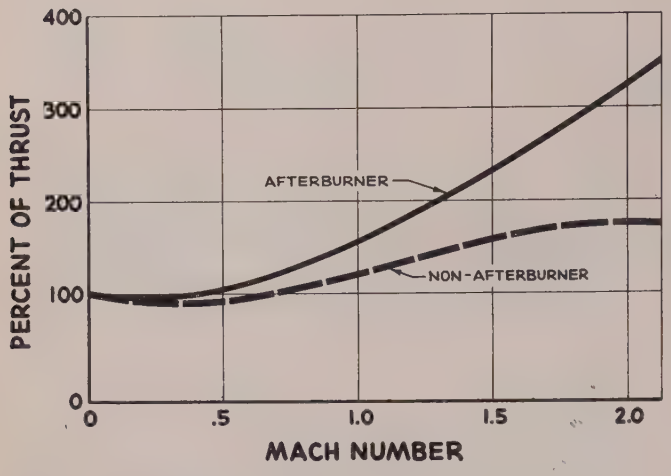

Fig. 1 – Thrust variation with Mach number for afterburner and non afterburner engines

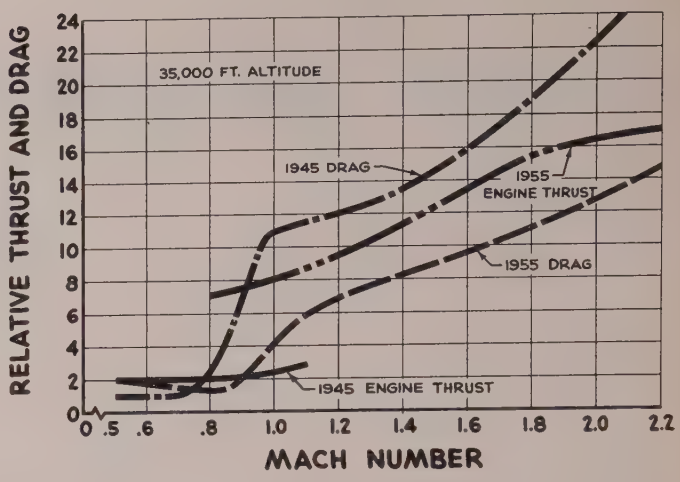

Fig. 2 – Comparison of thrust and drag in 1945 and 1955 fighter aircraft

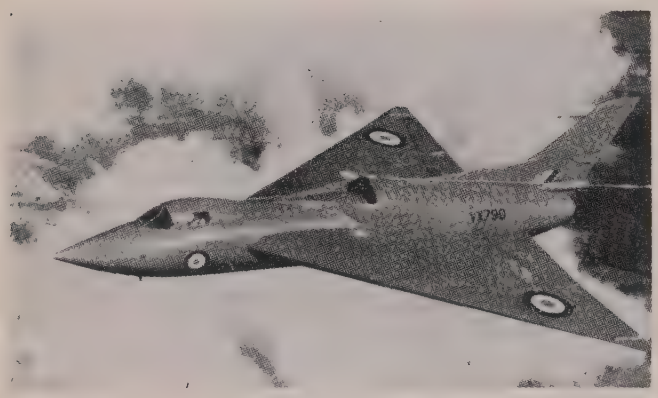

Fig. 3 – Avro 707A delta

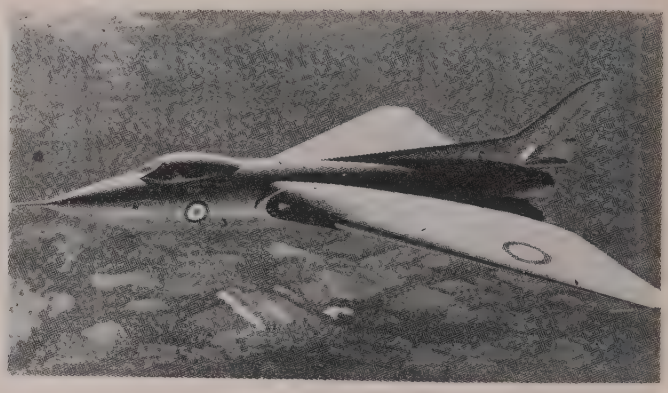

Fig. 4 – Avro 707B delta

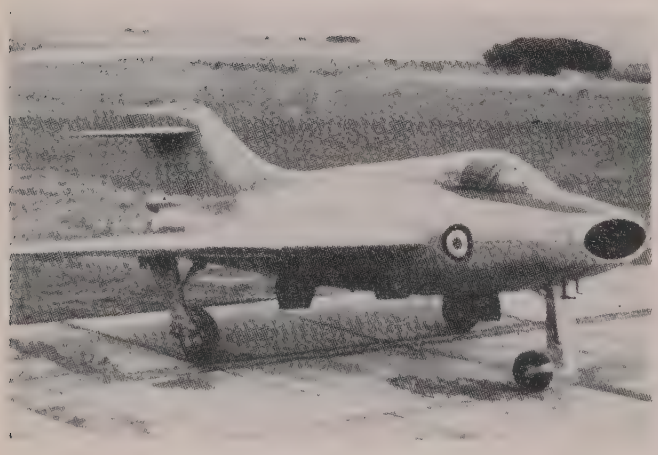

Fig. 5 – Boulton-Paul PX-120 delta

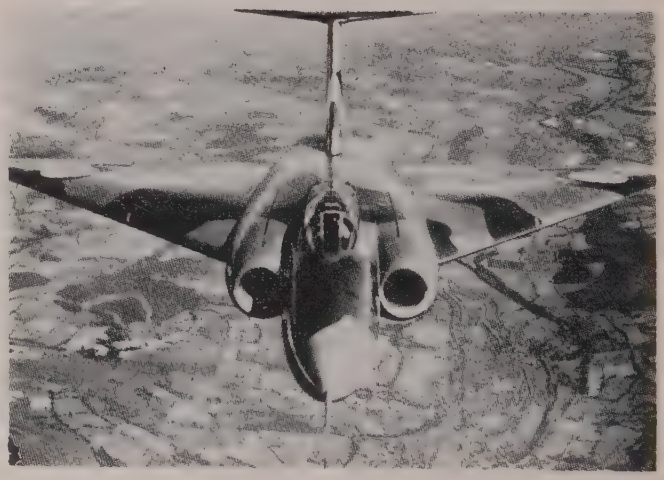

Fig. 6 – Gloster Javelin delta

formance superiority to the afterburner-equipped airplane. Also, the diameter of the afterburning engine is less than the diameter of the pure jet engine for the conditions noted, and the overall installed weight, including the added structure required for the airframe, is greater for the non-afterburning type by a substantial margin. The engine weights alone vary about 25% in favor of the afterburner type.

These are the basic reasons why the supersonic airplane will be equipped with afterburning rather than nonafterburning engines.

While the rocket powerplant is ideal for high performance for a few minutes, its fuel consumption, which is six to eight times that of the afterburning jet, rules it out for almost every reasonable mission in manned aircraft, except as an auxiliary device.

The plain ramjet cannot yet cover a wide enough speed range to be the basic airplane powerplant. Its best form, for some time to come, is as the afterburner behind the turbojet engine.

Fig. 2 shows an interesting comparison of the changes in thrust and drag which have taken place in the last 10 years. The Lockheed F-80 engine and airframe were designed in 1944. This airplane had a 13% thick wing, of aspect ratio 6.3. The installed engine thrust was approximately 4000 lb. Fig. 2 shows the drag of the F-80 configuration to Mach numbers of 2. There was no possibility of getting this airframe to fly supersonically with any air-burning engine it could carry. Now, 10 years later, engine thrusts for a similar size fighter have increased by 300%, and use of the afterburner gives increasing thrust with air speeds in the supersonic flight range. Likewise, by use of modern aerodynamic practices, airframe drag at Mach number 2 has been reduced to about half that of the older airframe. It is the combination of the thrust increases provided by the engine industry and the reduction of airframe drag on the part of the aircraft manufacturer that allows construction of practical supersonic aircraft.

One additional factor not generally known is that aircraft equipped with air-burning jet engines will, in almost every case, obtain their optimum speed performance at the isotropic altitude. This is normally at a height of 35,000 to 40,000 ft. Above this altitude, performance reduces rapidly, because the jet engine is no longer favored by dropping air temperatures, as it is from sea level to that height.

Let us now consider a few generalities of probable supersonic airframe types.

Discussion of Basic Types

The Delta Wing – There has been very limited experience obtained with delta-wing-type aircraft. This design, which was pioneered in Germany by Mr. Lippich, has several intriguing features. Figs. 3 to 9 show a number of types that have been built and flown in the United States and England. Varied degrees of success have been obtained with these

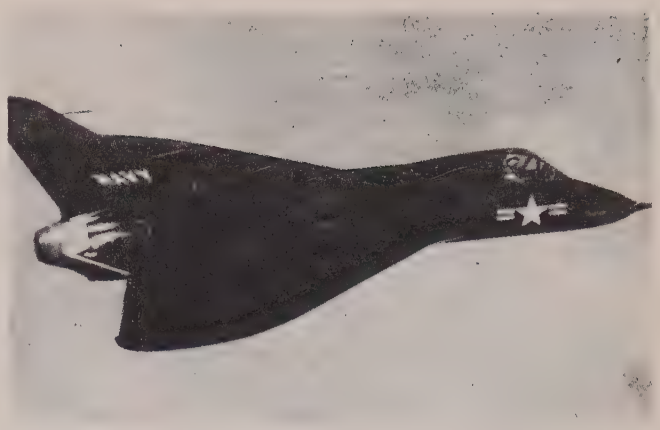

Fig. 7 – Navy's Douglas F-4D-1 delta

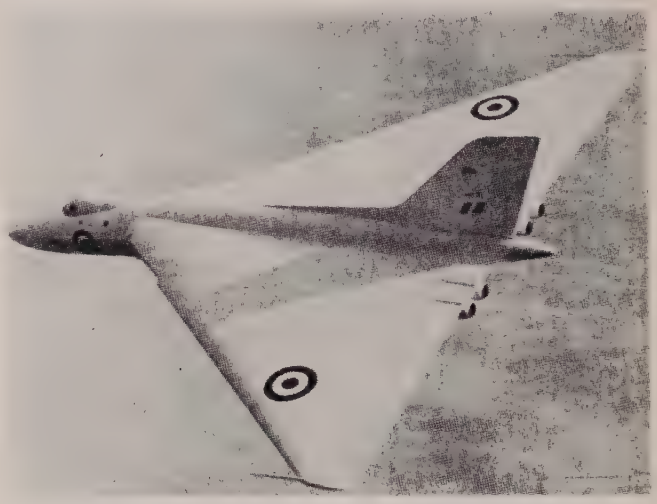

Fig. 8 – Avro 698 delta

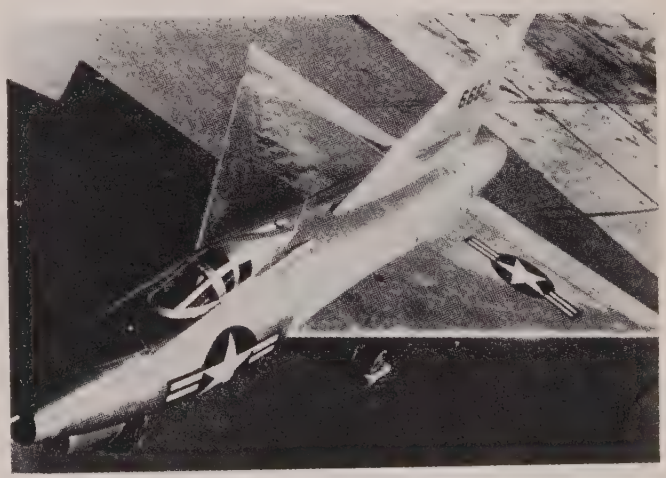

Fig. 9 – Convair XF-92A delta

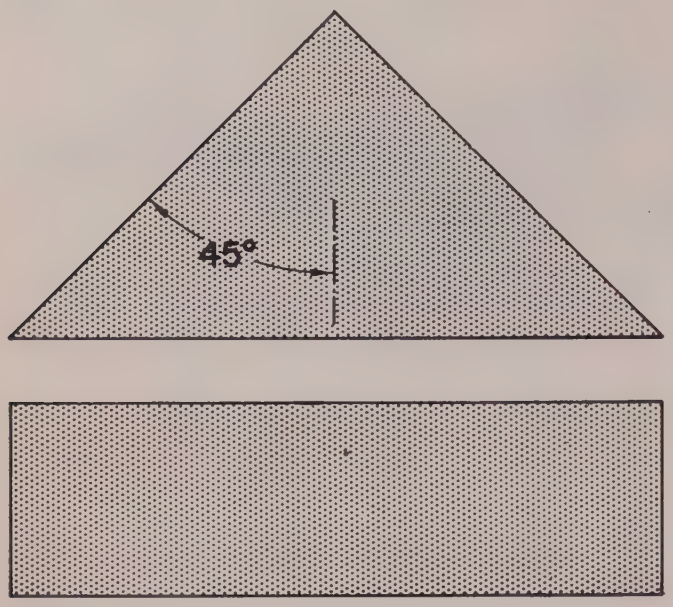

Fig. 10 – Comparable planforms – delta wing versus straight wing (aspect ratio 4.0)

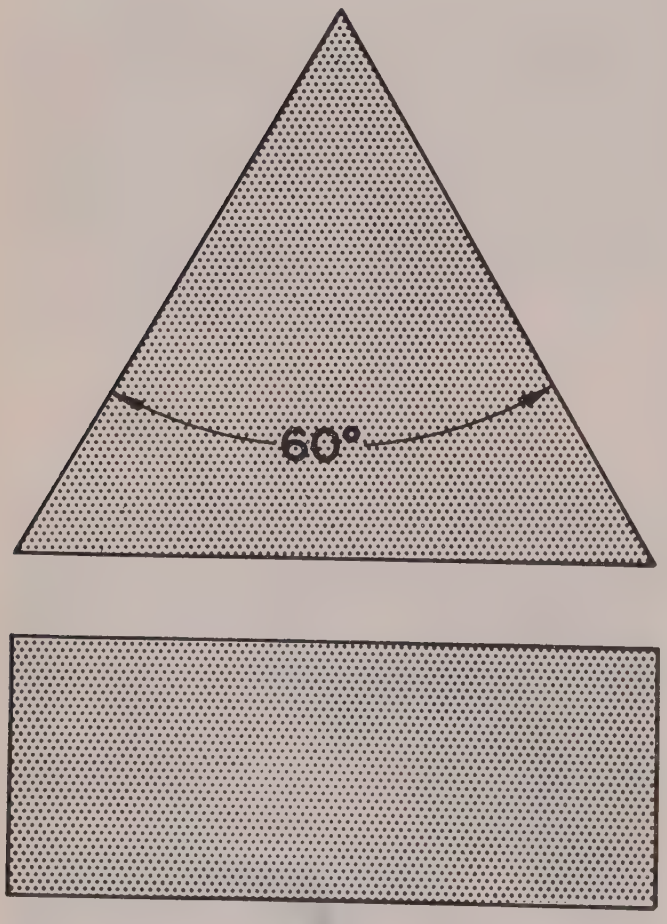

Fig. 11 – Comparable planforms – delta wing versus straight wing (aspect ratio 2.3)

Table 1 – Comparison of Surface Areas – Various Airplane Configurations

Airplane Model	Overall Length, ft	Wing Area		Horizontal Tail Area		Vertical Tail Area		Total Area of Wing and Empennage	
		sq ft	% of wing area	sq ft	% of wing area	sq ft	% of wing area	sq ft	% of wing area
F-80A	34.5	237.6	100	43.5	18.3	22.5	9.5	303.6	127.8
F-86	37.54	287.9	100	33.9	11.8	33.4	11.6	355.3	123.4
XF-92	42.5	425	100	—	—	76.1	17.9	501.1	117.9
F-94C	40.08	232.8	100	60.3	25.9	28.0	12.02	321.1	138.0
F7U-3	43.17	535.3	100	—	—	147.4	27.5	682.7	127.5
Avro 698	109	4230	100	—	—	294	6.9	4524.0	107.0
Avro 707A	34.17	426.5	100	—	—	29.9	7.0	456.4	107.0
Gloster Javelin	57.0	905	100	79	8.7	61	6.7	1045.0	115.5
Boulton-Paul PX-120	29.63	—	—	—	—	—	—	—	—
Boulton-Paul P-111	26.1	291.0	100	—	—	33.5	11.5	324.5	111.5

airplanes, which have been flown almost entirely in the subsonic speed range. No delta-wing airplane has obtained the high speeds reached by rocket-powered straight- or swept-wing research aircraft. In fact, few have reached speeds obtained by service aircraft. Within the next few years, this situation will, no doubt, be changed.

The advantages of the delta wing are primarily those having to do with structure. The high taper ratio tends to reduce the number of structural problems, although, as one gets further into the designs, there are many important structural problems which still exist. The delta-wing airplane without a horizontal tail must be predicated to fly at low angles of attack because of the high drag increase that comes about at high lifts. The longitudinal control is obtained by turning up elevons on the wing trailing edge, which decrease the lift considerably, in order to obtain pitching control. This feature, together with the high leading edge sweep, results in very low maximum lift coefficients for the delta configuration, which is the chief drawback of the type. Because the triangular wing section results automatically in low aspect ratio for any practical leading edge sweep, many of the structural advantages that go with low aspect ratio have been credited to the delta. Figs. 10 and 11 compare the 45- and 60-deg delta aircoils with rectangular planforms of similar aspect ratio. After years of drawing wings in the aspect ratio range of from 6 to 12, it is hard for a designer to bring himself to draw those of 2 and 3 in rectangular planform. When he does, it becomes immediately apparent that the delta airplane has no patent on the structural advantages that go with low aspect ratio.

The delta airplanes, in the main, have been the so-called tailless type. This is a very considerable misnomer because, while they may not have horizontal tails, they make up for it by requiring terrific vertical tails. Table 1 and Fig. 12 show a few

comparisons in terms of wing area and total area, including horizontal and vertical tails, of several nonclassified aircraft. It is immediately apparent that many of the delta-wing types have as much combined surface area, without horizontal tails, as conventional aircraft which have such surfaces. Also, while the comparison is very rough, the low wing loading of the delta types required to get reasonable take-off and landing distances actually results in higher surface area per pound of gross weight than with conventional types. For supersonic aircraft, this is a very undesirable trend.

When the delta-type airplane is equipped with a horizontal tail with some reasonable tail length, higher wing loading can be carried and the overall delta performance improved. Fig. 13 shows a typical design for the tailed delta type and Fig. 14 an unswept straight-wing type.

To fly at supersonic speeds requires wing thicknesses that are so low that one of the primary advantages previously credited to the delta airplane has disappeared. This was the ability to put a great deal of fuel or equipment inside the wing. Actual delta designs are showing that the fuselage cross-section and length (see Table 1) are as great for the delta type as for the conventional airplane. It is impossible to reduce the cross-section of the engine, the military equipment, the air ducts, or the pilot, by choice of wing planform, to any important degree. This means, essentially, that the aerodynamic drag of the delta type becomes greater than that of other types at supersonic speeds, because, while fuselage drags are similar, the lower wing loading of the delta and its high vertical tail area increase its total drag. A major error in evaluating the delta type is generally made in considering wing area without considering maximum lift and drag at high angles of attack. While the pounds per square foot sound good falling off the slide rule, the wing doesn't show up very well with its limited pitch control and low usable maximum lifts on take-off and landing.

The major advantages of the delta wing are in its structural rigidity and in the improved space generally available in the type.

The Lockheed Delta Design – A number of years ago, Lockheed undertook intensively the design of a delta-wing airplane, with the results shown in Fig. 15. This airplane actually progressed to the point where pieces of an airplane were being built in the shop. Extensive tunnel tests, however, showed so many aerodynamic problems connected with the type that it was discarded. One of the major problems encountered was the directional instability at high angles of attack. Fig. 16 shows the variation of the directional stability with two different vertical tail configurations. The reversal of directional stability prior to the wing stall was considered to be an extremely dangerous condition. Use of a double tail was beneficial at some angles of attack but did not correct the situation. Comparison of the delta configuration with a con-

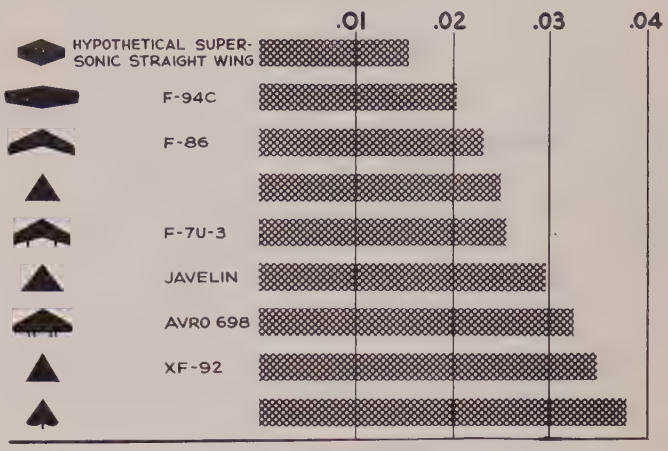

Fig. 12 – Total area per pound of gross weight

Fig. 13 – Delta-wing fighter with conventional tail

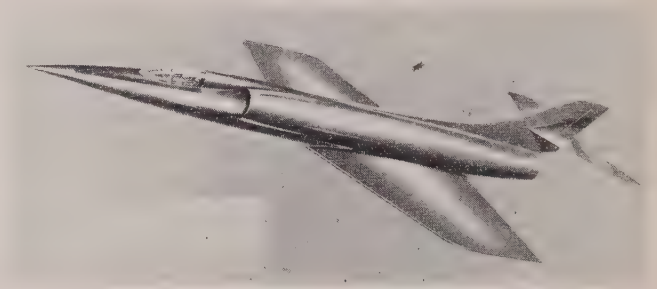

Fig. 14 – Low-aspect-ratio straight-wing fighter

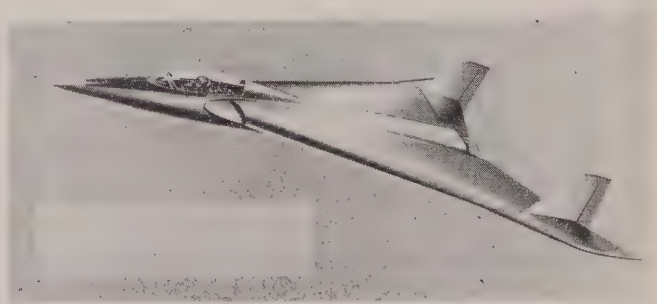

Fig. 15 – Delta-wing fighter – Lockheed 1947 design study

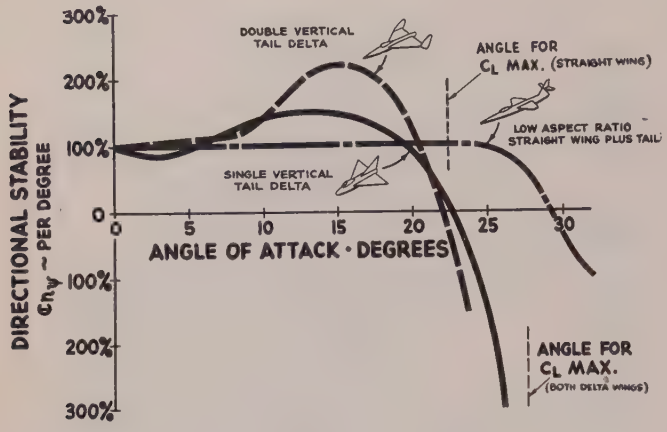

Fig. 16 – Directional stability versus angle of attack

ventional low-aspect-ratio straight-wing airplane shows the effect of wing planform on this problem. The delta-wing airplanes had reverse stability *prior to* the stall. The straight-wing type got to zero stability considerably *above* the wing stall.

A second major problem encountered was the great loss of lift encountered when the wing was trimmed for any given angle of attack. Fig. 17 shows the maximum lift obtained both with and without trim on the airplane with the landing gear extended. Numerous efforts were made to trim the airplane by trailing edge devices, leading edge devices, extendable horizontal surfaces on the nose of the fuselage, and, finally, by wing tip trimmers. By moving the whole outboard wing section beyond the vertical tails to angles of attack as much as –55 deg, suitable trim characteristics could be obtained. Devices on the forward part of the airplane which showed favorable trimming characteristics

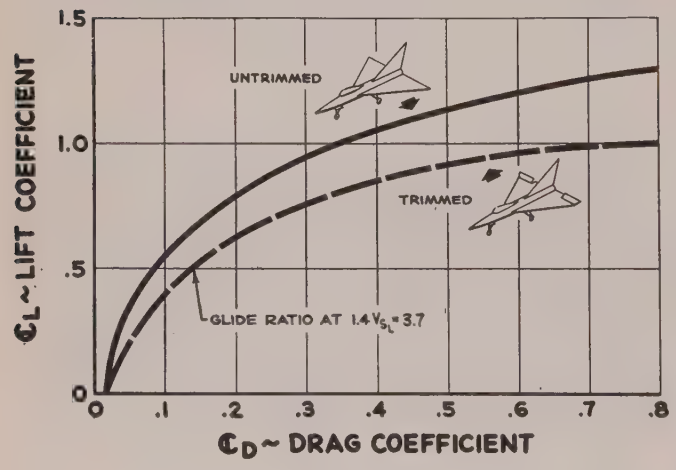

Fig. 17 – Trim drag of tailless 60-deg delta, landing configuration – gear down

always ruined the stability. Because of the combination of low aspect ratio and loss of lift due to trim, the lift-drag ratio when gliding at a speed of 40% above the stall was 3.7 to 1. Values for our current airplanes, such as an F-80, at similar glide speed are approximately twice as high (Fig. 18). To get the same type of landing flare characteristics as were available on the F-80, a thrust output equal to normal power on one of the two delta-wing airplane engines was required.

Landing flare rates of descent required very long landing-gear shock-strut travel. The final value chosen for the Lockheed design in order to get reasonable load factors was 25 in.!

While the basic wing structure started out to be quite light, two factors came into the picture which were very adverse. These were the high suction forces acting over large parts of the wing in pull-up maneuvers at low altitude and the use of numerous stressed cover plates to provide access to equipment in the wing, to make use of space that was available.

When the actual weights were computed for the final design, the delta-wing-type airplane weighed a few hundred pounds *more* than a competitive design utilizing a normal swept-wing design with a tail.

A final evaluation of drag through the transonic range, comparing the Lockheed delta to other wing configurations, led to the discarding of the projected airplane.

Swept-Wing Types – The swept-wing planform has demonstrated good success in the high subsonic and transonic speed ranges. As the sweep angle of the leading edge is increased for high supersonic speeds, a number of undesirable characteristics of the swept wing become apparent. Structurally, the swept-wing airplane must also use low aspect ratios. For a given aerodynamic aspect ratio, however, the structural aspect ratio remains high. This gives high wing and fuselage design weights and a considerable number of aeroelastic problems. The maximum lift available is poor, and the problems of pitch-up near the stall become extremely difficult. It is not believed that for high supersonic speeds the purely swept wing will be a contender with the straight and delta types.

Straight Wing – The wing type referred to as "straight" in this paper has essentially zero sweep to a line between 25 and 70% of the wing chord. The wing may be tapered, and probably must be, for structural reasons. If we assume that the low-aspect-ratio characteristics of the delta airplane will be acceptable to pilots, we can use similar low aspect ratios with the straight-wing types.

A number of papers have been published on wing planforms wherein straight wings of high aspect ratio, requiring wing thicknesses of 10 or 12%, have been compared with low-aspect-ratio delta types with wing thicknesses of 5%. The delta-wing airplanes win every time with such a comparison. If we do accept the low aspect ratio for the delta

and then screw up our courage to draw a comparable straight-wing planform, we then find it possible to obtain comparable wing thicknesses in terms of per cent chord. The major problem that faces us in this condition is a problem of aeroelasticity and flutter. Even these problems have been proven capable of solution. The straight-wing airplane has fundamentally better landing and take-off characteristics than the delta or swept type. This is because, even in the extremely low thicknesses, it is possible to obtain lift coefficients very substantially higher than those of the delta. Because of the higher maximum lift, it is possible to use much higher wing loadings for given take-off and landing performance.

There are many ways to compare airplanes in the theoretical design stages. However, as long as our aircraft run along the ground before and after getting into the air, there is an inevitable real estate problem which is always a great leveling factor. If we are going to compare airplanes, let's compare them on the basic criteria that the field lengths are the same, regardless of wing planforms. Let's further compare them in that they must do the same mission, with the same equipment or payload. The real estate problem is the one most generally neglected when discussing delta-wing airplanes. Exponents of the type keep discussing low wing loading, but they almost always forget that the low maximum lift the type develops largely counteracts most benefits from the slide rule number.

The straight-wing airplane has generally the same type of stability characteristics we have become familiar with over a number of years. All low-aspect-ratio aircraft, however, encounter certain problems of lateral-directional stability which will, no doubt, require artificial stability devices. The straight-wing airplane will have its basic stability advantages over the delta at high angles of attack, while having similar stability at high speed. Several of these factors will be discussed at greater length in a later section of this paper.

Let us consider a number of comparisons of specific data, to illustrate some of the foregoing points.

Drag Factors

It is interesting to consider the change in the drag breakdown of a typical tailed fighter aircraft when going from subsonic to supersonic flight. Fig. 19 gives a comparison of this nature. Doubling the Mach number increases the drag to 520% of its value at a Mach number of 0.80. The wave drag becomes an extremely important quantity. The pitching moments of the wing and the horizontal tail itself cause a measurable amount of trim drag.

The extremely important effect of wing thickness on the wing drag is shown in Fig. 20. At a Mach number of 1.5, the 3%-thick wing has only 63% of the drag of a 5%-thick wing of the same planform. Subsonically, the thinner wing develops leading edge flow separation when no leading edge droop is

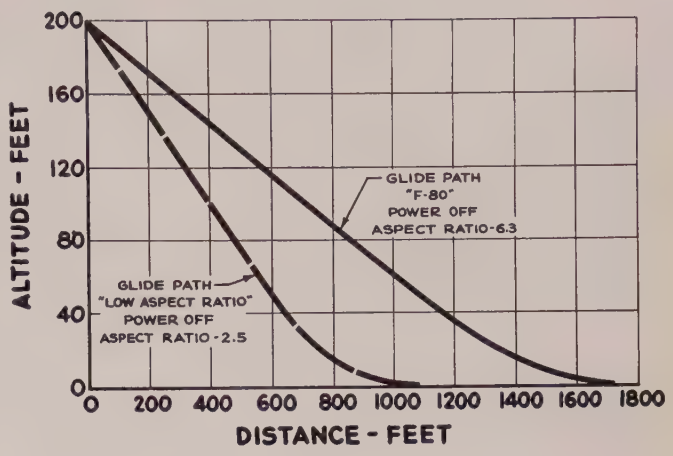

Fig. 18 – Glide-angle comparison – low aspect ratio versus F-80

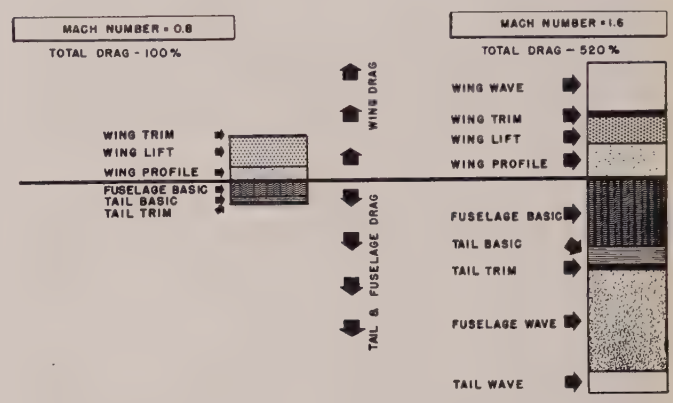

Fig. 19 – Comparison of drag components for a fighter aircraft at two Mach numbers at 35,000 ft

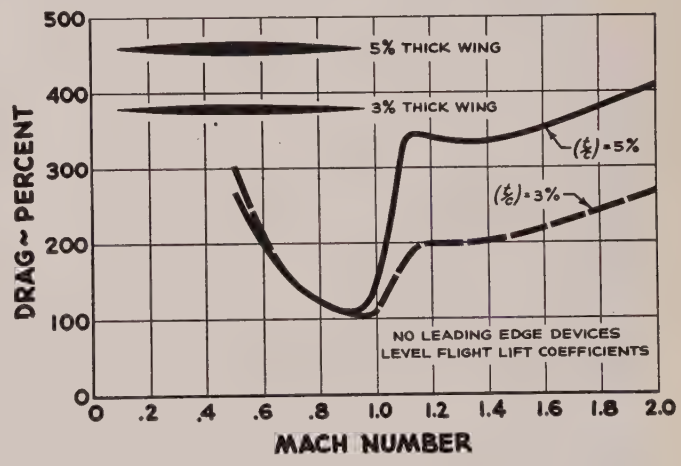

Fig. 20 – Effect of thickness on drag for straight wing alone

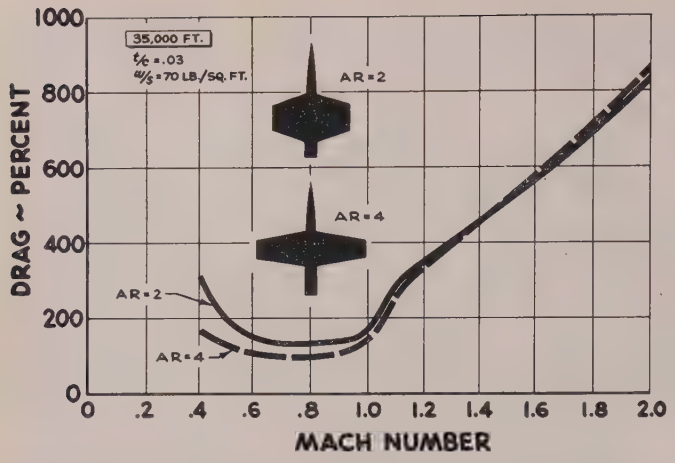

Fig. 21 – Effect of aspect ratio on drag for straight wing plus fuselage

employed, so its drag becomes higher than for the thicker section.

In this same figure are shown the actual cross-sections of a 3%- and 5%-thick wing. Considering the structural problems, it is easy to see why no important space remains in the wing for fuel, landing gears, or equipment.

The low-aspect-ratio, thin wings seldom show a constant overall airplane efficiency factor *"e"* on induced drag, due to leading edge flow separation and other causes. This condition which was sometimes assumed to disappear with increased Reynold's number has not done so. Many early design studies missed this point and, therefore, developed substantial errors. Leading edge flaps help the unswept wing *"e"* factor, but show reluctance to do much with the higher sweep angles.

The drag due to lift supersonically does not appear to vary much for different planforms of comparable aspect ratios.

The effect of aspect ratio on airplane drag is illustrated in Fig. 21 for two wings of 3% thickness. The lower aspect ratio has higher drag over most of the important speed range, particularly in the take-off and low-speed range. There is nothing good aerodynamically about low aspect ratio, which, of course, we have known for years. It is required for high-speed aircraft in order to allow low thickness ratios which would be structurally impossible with greater spans. The data are shown for the wing with a fuselage because so much wing area gets buried in the fuselage with the lower aspect ratios that this factor must be accounted for.

The tailless delta airplane suffers considerably in its maximum lift-drag (L/D) ratio by requiring an upturning trailing edge flap for longitudinal control. Tailed aircraft also have important trim drag, but our experience shows it to be less than for the delta type. Figs. 22 and 23 show the effect of trimming a typical delta aircraft by a trailing edge flap and compare the L/D's of two types of aircraft. At high speeds, the L/D ratio with trim is only 62% of the untrimmed value. Compared to a straight-wing design of comparable aspect ratio and thickness ratio, the delta-wing aircraft shows a lower L/D ratio, both subsonically and supersonically.

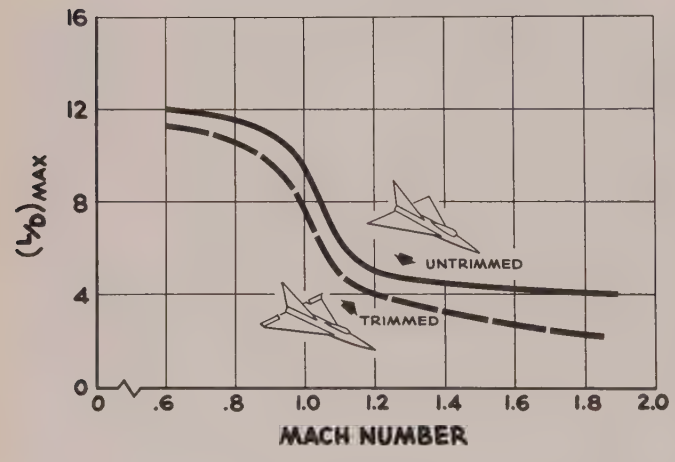

Fig. 22 – Effect of trim drag on maximum L/D versus Mach number for tailless delta

Lift Considerations

The maximum lift limitations of the delta-wing aircraft without a horizontal trimmer are due in large degree to the lack of a down-turning trailing edge flap. In fact, the conventional trailing edge trimmer turns upward to reduce the maximum lift greatly. To date, little success has been obtained in trimming the delta wing with leading edge devices. Nor do most of them add much to the maximum lift.

The maximum lift on the pure delta type is obtained at such a high angle of attack that it is unusable with practical ground angles and landing gear lengths. Use of a trailing edge lift flap (instead of the antilift trimmer) corrects this condition by shifting the zero lift angle substantially.

If a conventional horizontal tail is put on the delta airplane, a greatly improved situation results. A moderate amount of wing flap can now be used on the delta type, and a much better aircraft results. The maximum *usable* lift for take-off and landing improves by 45%, and the higher wing loadings made possible considerably improve the overall performance, including level flight speeds.

The conventional straight-wing airplane is the best for obtaining optimum usable lift. Leading edge devices are extremely effective, as well as trailing edge lift flaps. For a given wing area, more wing is outside the fuselage to develop lift and subject to being affected by lift devices. The

shorter aerodynamic chord reduces the down tail load required to trim. Fig. 24 summarizes the comparative lifts obtainable for the three types. If the maximum *usable* lift of the straight wing is taken as 100%, the tailed delta develops 70% and the pure delta only 48% as much.

When these values are used to design aircraft with comparable take-off and landing performance, it is obvious which type gives the smallest, lightest, highest performance type. Even if a considerable sacrifice is made in field lengths to favor the delta, the straight-wing type is markedly superior.

Stability and Control

The use of low aspect ratios and high sweeps, together with long, dense fuselages, gives all forms of supersonic aircraft problems in stability. Yaw and roll dampers will probably become standard equipment. Flight characteristics at low angles of attack will be quite reasonable on all types, but at high angles many problems still exist. Some types will and do exhibit tendencies to lose or reverse all directional stability and, therefore, "swap ends" at high angles of attack. Some will have high rates of Dutch roll just above take-off speeds. The longitudinal axis will have its full share of problems.

The transonic trim changes which plague us today will automatically disappear with reduced wing thickness and better aerodynamic forms. Any airframe which has low enough drag to fly supersonically in level flight will also have greatly improved transonic flight characteristics over our present aircraft.

The buffet characteristics of thin wings at high lift are largely unknown. We may have to accept some transonic buffeting.

The fuselage is playing an increasingly important part in determining airplane flight characteristics. The high fineness ratios required and the relatively large dimensions of the fuselage cross-section as compared to the wing span markedly affect the high angle of attack flow conditions over the wing and tail.

Certain claims have been made by delta-wing enthusiasts that much less aerodynamic center travel is obtained transonically than for other wing planforms. In terms of per cent chord this is certainly true – it's half as much as for a straight wing. However, if put in terms of inches of travel, which is the real criteria, the center-of-pressure travels are almost identical for the two wings. The very restricted allowable center-of-gravity range for the tailless delta makes the trim change problem much more serious for this type than others.

Overall, our tests have shown the straight-wing aircraft with proper tail design to have stability and control characteristics superior to those of other forms of wing planforms.

Other Factors

Most of our large subsonic bombers and jet transports appear to be using the swept, relatively

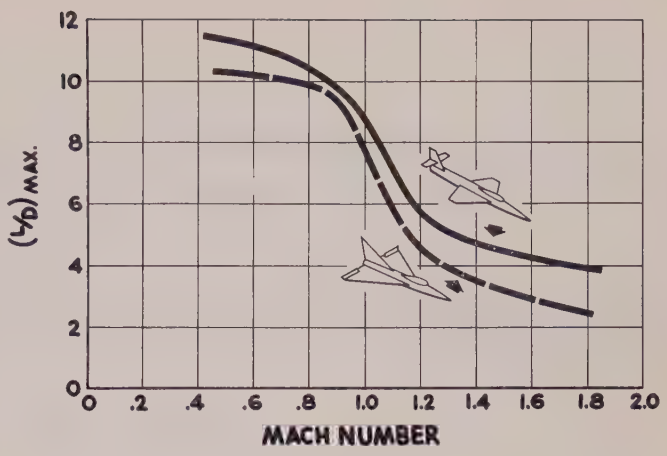

Fig. 23 – Maximum L/D versus Mach number for trimmed flight – typical delta-wing versus straight-wing airplane with same wing thickness ratio

thick wing types. For Mach ranges up to 1.0, this type is an optimum design, mainly due to the requirement for great fuel volume, which is sacrificed if an equivalent drag thin wing is used. For supersonic flight, this consideration must be discarded.

The subsonic delta wing, with all its problems of control and low aspect ratio, does not appear comparable to the swept wing in this speed range when evaluated for the *same mission* and with the same powerplants.

Conclusions

With the solution of the aeroelastic problems associated with the unswept, very thin wing, this planform appears to have substantial advantages over other types when evaluated on a strictly comparative basis for a flight regime which reaches well into supersonic speeds.

DISCUSSION ON FOLLOWING PAGE ——→

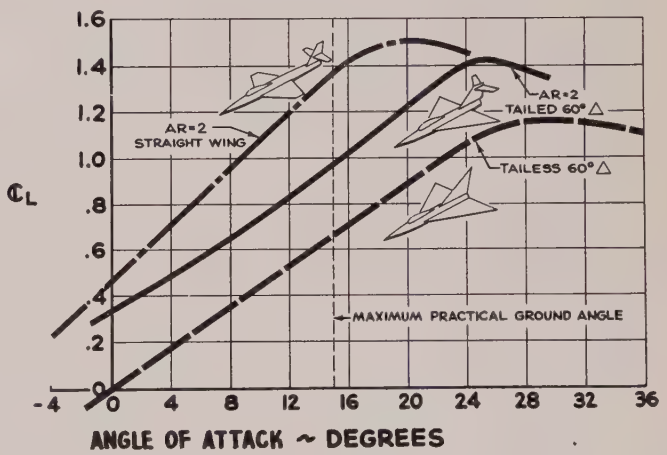

Fig. 24 – Maximum lift comparison – landing configuration

DISCUSSION

Presents the Case Defending Deltas

– Adolph Burstien
Consolidated Vultee Aircraft Corp.

WE consider Mr. Johnson's paper both timely and provocative. However, we wish to discuss some of the points brought out by Mr. Johnson, within the limits of security.

We certainly agree that styles in engineering are bad, nor do we wish to defend deltas featured in advertising, since we have been too busy to scrutinize these cartoons. Mr. Johnson is also correct in stating that no full-scale rocket-powered delta research aircraft had been flown at extreme speeds and extreme altitudes. However, with the powers and altitudes used, almost anything could be made to go fast. The recent speed record flight of the Douglas F-4D may be more to the point in illustrating capabilities of a configuration that resembles a delta.

This paper gives one the impression that Mr. Johnson's opinions are based primarily on data of 1947 vintage. We all know that very extensive study and tests of all configurations by NACA and others have taken place since that time.

Looking at Figs. 10 and 11, we draw the opposite conclusion from the author as to structural advantages of deltas. If we consider bending at the side of a typical fuselage supported by these wings, it looks like only 40 to 50% of bending material would be required for the delta wing of the same per cent thickness. The torsional stiffness of the delta would be about eight times that of the straight wing at the inboard end.

Table 1 was used to show that deltas require very large vertical surfaces in terms of wing area. However, the British deltas listed have proportionately smaller vertical surfaces than the F-80 or F-86. We can only apologize for the large vertical of the XF-92 and blame it on undue conservatism. We are not repeating this conservatism in our subsequent designs.

Mr. Johnson points out correctly that modern fighter fuselages are tightly packed with engines and equipment. However, he is incorrect in saying that wing volume of a thin delta is unusable for fuel. One thing that puzzles us is where does he carry his fuel and landing gear? The implication is that they are neither in the fuselage nor in the wing.

A big point has been made of the so-called low maximum C_L available in a delta airplane. As we all know, other factors besides landing speed, such as ceiling, maneuverability, and available volume, must be considered in choosing wing area. We have considered putting horizontal tails on our deltas, but, after evaluating all the factors, decided against them. We find that areas chosen on the basis of considerably less than maximum C_L do not penalize these other considerations. And this is done without the help of droop snoots, flaps, tails, or boundary layer control. Therefore, the C_L's we use are in a clean configuration which certainly helps the high-altitude maneuvering requirements.

Data similar to that of Fig. 16 had given us some uneasy moments at first, even though the data shown is unnecessarily pessimistic. However, two factors must be considered: As already stated maximum C_L is not utilized, and the XF-92 has been flown full scale and in the free-flight tunnel at angles of attack well beyond the zero stability point without difficulty.

A big issue is being made of the lift and drag penalties associated with trim of tailless deltas. One factor not generally realized, but pointed out by Mr. Johnson, is that fuselage lengths of comparable aircraft are about equal for all wing configurations. This means that geometric control lengths are in the same "ball park." This is particularly true for the supersonic case, which is the most critical. As a matter of fact, we have always considered the maneuverability of a delta as one of its important advantages.

Some very sad L/D's have been quoted, evidently based on Lockheed's 1947 study. Unfortunately, security prevents us from discussing this subject fully. However, certain physical factors and tricks learned by NACA and ourselves paint a much more cheerful picture. As a matter of fact, the L/D quoted in the paper for the approach condition is actually at least twice as much, or similar to the F-80. Lockheed's study indicated that 25-in. shock-strut travel is required for landing. FJ-1 shock struts are used on the XF-92 airplane satisfactorily and represent conventional shock-strut travel.

The real estate problem mentioned by Mr. Johnson is of real importance. However, probably as many Convair engineers as Lockheed engineers own property, so that we, too, are aware of its value. Take-off distance is a function of take-off C_L, wing area, drag, and thrust. We have found that, for our designs, take-off distance is no problem once sufficient thrust is available for high-speed flight. As previously mentioned we have offset somewhat lower C_L by larger wing area, which did not appear to jeopardize the airplane mission in any way.

Figs. 22 and 23 were used to strengthen further the case against the tailless delta. Security prevents us from quoting actual points, but we can say that we have succeeded in designing much more efficient aircraft than indicated.

Many of the items shown in Fig. 24 were already discussed. However, we do not agree that 15 deg is the maximum practical ground angle. We, and others, have been able to design for and use considerably higher landing angles.

The statement that the tailless delta is very seriously restricted in allowable center of gravity range is incorrect. The usable c.g. range in inches has been very comparable to other configurations, and no unusual airplane layout problems have been encountered.

In conclusion, these remarks were not intended to induce Lockheed to change to delta but to correct some of the statements made in the light of our experience. Convair does not claim that the delta wing is a panacea for all aircraft designs and problems. For some of our designs delta came out to be the best solution.

Author's Closure To Discussion

WHILE some of my comments regarding delta wings were based on research on an original Lockheed design, by far the major portion of the information shown is of much more recent date.

In answer to his question on where the fuel and landing gear are carried, it is obvious they must both go in the fuselage.

I believe Dr. Burstien does not disagree that considerably more wing area is needed for the delta type to get equivalent takeoff and landing performance. This is the very factor that penalizes its supersonic performance, so apparently we have no argument on that score.

Tractor Hitches and Hydraulic Systems— An Implement Designer's Viewpoint

B. B. Johannsen, John Deere Plow Works, Deere & Co.

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IMPLEMENT hitches and hydraulic systems provided with present-day agricultural tractors are in general designed by the tractor manufacturer's engineering group. These designs usually represent the tractor designer's idea as to what is required to operate a variety of implements properly, and such design may not necessarily be the best to operate successfully a particular type of implement produced by an implement manufacturer.

Transferring the tractor-hitch design to the implement designer will also present problems in that the implement designer is not entirely familiar with tractor limitations; and further, even though the men in this design group are familiar with the

SINCE World War II, integral implements have become increasingly popular. According to the author, this ever-increasing popularity of integral instruments over the corresponding towed type can be attributed to the following factors:

1. Greater maneuverability of tractor and implement combination.
2. Better transporting characteristics of tractor and implement combination.
3. Lower-cost implements.
4. Ease of attachment and control.

An implement designer's viewpoint on tractor hitches and hydraulic systems is presented here. The author hopes that it will help to assure that tractors of the future will have common implement-to-hitch attaching points, so that a tractor may be used with a number of different makes and sizes of integral implements.

He also points out that certain performance features are needed to make integral instruments even more popular. These include: provision for their use with competitive tractors, quick and easy attachment of implements, simple adjustments, provision for leveling, adequate hydraulic system lifting capacity, means for depth control, provision for remote cylinder, and satisfactory tractor longitudinal stability.

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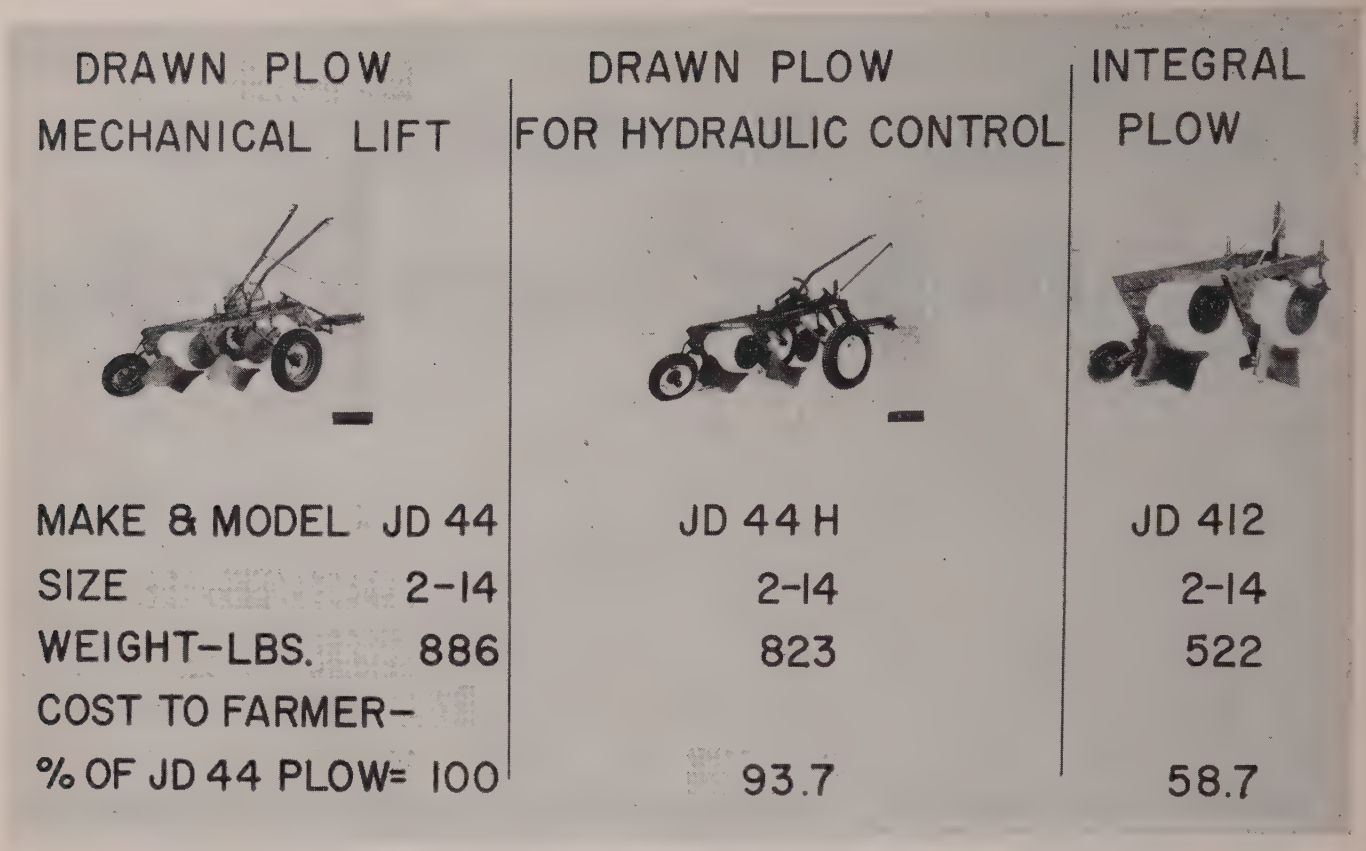

Fig. 1 – Comparison of integral versus drawn 2-bottom plows

specific requirements of their own particular implements, they do not always fully understand the problem peculiar to other implements. In present-day circumstances, farmer customers usually expect one make of implement to function successfully with a number of different makes and models of tractors or vice versa. It therefore appears that a satisfactory design of hitch and hydraulic system must be reached as a result of close cooperation between the tractor and implement designers so that individual design problems and limitations may be mutually understood.

The purpose of this paper is to present the implement designer's problems in the hope that future tractors will have hitches and hydraulic systems which will more nearly fit the field requirements.

Why Integral Instruments Are Popular

Since World War II, integral implements have become increasingly popular. This ever-increasing popularity of integral implements over the corresponding towed type can be attributed to the following factors:

A. *Greater Maneuverability of Tractor and Implement Combination* – With the use of integral implements, the farmer can successfully work small or irregularly shaped fields that are often impossible or impractical to work with a drawn implement. Integral tillage tools, planters, and cultivators can be readily lifted and backed into corners or hard-to-reach places, whereas it is rather difficult to back up with drawn equipment. This also applies to scoops, blades, and like implements when used for moving materials and making fills around farm buildings.

B. *Better Transporting Characteristics of Tractor and Implement Combination* – In the past 12 years, there has been a great influx of farm labor to the cities to work in industrial and defense plants. This has resulted in farmers now farming more land as well as farming land a greater distance from their homes. Modern tractors, equipped with rubber tires and transport speeds of 10 to 20 mph, take care of this circumstance. Customers, therefore, desire integral implements that can be transported safely at the higher transport speed of the modern tractor. It is also recognized that the number of hard-surface roads over which farmers must transport implements are steadily increasing. Generally, it is essential that the implement be kept from contact with road surface to prevent damage to the implement, the road surface, or both.

C. *Lower-Cost Implements* – Integrally mounted implements can normally be sold at lower cost because wheels, raising and lowering mechanisms,

lateral and longitudinal adjustments, as well as depth-gaging means, are eliminated from the cost of each individual implement since they are furnished as a part of the tractor. The cost, when applied to the tractor, is therefore spread over a number of implements. Fig. 1 illustrates the comparative weight and cost of a drawn moldboard plow with mechanical lift, a hydraulically controlled drawn plow, and an integral plow each in the 2-bottom 14-in. size. Basing the cost of the 2-bottom drawn plow with mechanical lift at 100%, the retail cost of a hydraulically controlled towed plow is 93.7%, and the integral plow is 58.7% of the mechanical lift plow. In times of a buyer's market, the lower price tags on the integral implements are attracting more and more attention.

D. *Ease of Attachment and Control* – Integral implements lend themselves to a design such that the implement can be readily attached or detached, in many cases, without the necessity of leaving the tractor seat. Such implements are readily adjusted for longitudinal and lateral leveling from the tractor seat.

Although the features listed above have made integral implements popular, it must be recognized that there are limits to which integral implements can be used and that drawn implements will continue to be in demand. A drawn implement is universally adaptable to all makes and models of tractors equipped with a drawbar and a suitable hydraulic system for operating a remote cylinder on the implement. The single point of attachment consisting of a clevis and a pin makes it easy to attach or detach most drawn implements. Optimum-size drawn implements can utilize the full tractor power. This is particularly true for such implements as disc harrows, spike tooth harrows, rotary hoes, disc tillers, and so forth. If soil penetration is a problem, added ballast can be applied to a drawn tool, whereas such practice is often prohibitive on integral tools because of limitations of the tractor's hydraulic lift capacity or the possibility of adversely affecting the tractor's stability. In the wheat-growing states, multiple drawn tool operation is popular as it permits full utilization of both available time and labor. Tools commonly used in multiple hookup are tool-bar cultivators and rod weeders, grain drills and rod weeders, rod weeders and drag harrows, and tool-bar cultivators with drag harrows. With such multiple tool arrangement, farmers are able to farm more acres with less manpower, but they are not feasible with integral tools as we know them.

It is also recognized that drawn implements are not thrown out of adjustment when they are attached to or detached from a tractor. Untrained tractor operators, frequently employed by farms, ranches, and plantations, can successfully operate a variety of drawn tools interchangeably after each tool is once properly adjusted for the field condition. On the other hand, integral tools normally require separate hitch adjustment for each type of implement for satisfactory performance. To adjust an integral tool properly each time it is attached to a tractor is, in many cases, beyond the scope of an untrained tractor operator, therefore requiring the close attention of a field supervisor. To place these adjustments in each implement would probably inflict a cost penalty that could not be tolerated.

The popular performance features that have been incorporated in drawn tools throughout the years will continue as popular features in integral tools. Although in some cases the customer has demonstrated a willingness to sacrifice somewhat in the quality of the work done to obtain the advantages of integral tools, this sacrifice is made not because he likes to make it but because in many cases he is obliged to yield an advantage in order to gain other advantages integral tools might offer.

Performance Features

The following list of performance features should therefore be given consideration in the design of

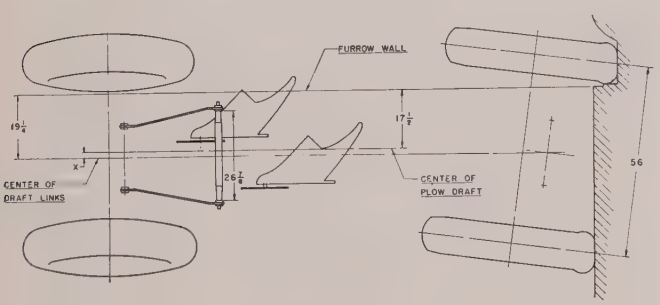

Fig. 2 – Implement and tractor alignment – 2-bottom, 14-in. plow, 26⅞ crossbar, 10-in. tires at 56-in. tread

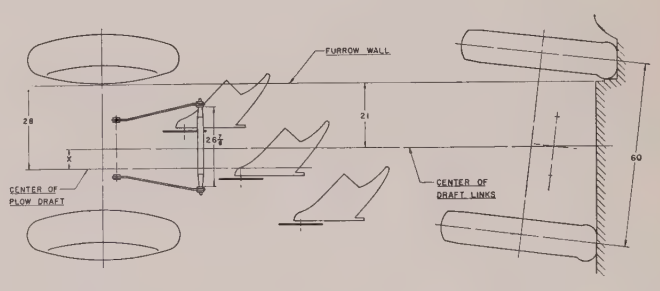

Fig. 3 – Implement and tractor alignment – 3-bottom, 16-in. plow, 26⅞ cross bar, 10-in. tires at 60-in tread

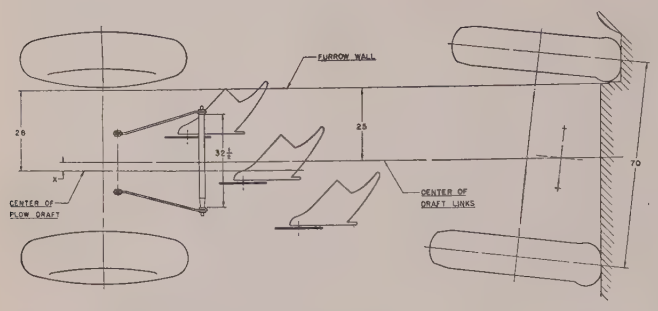

Fig. 4 – Implement and tractor alignment – 3-bottom, 16-in. plow, 32½ crossbar, 12-in. tires at 70-in. tread

the implement hitch and hydraulic system in order to make integral equipment more popular or more readily acceptable to the consumer.

A. *Provision for Implements to be used with Competitive Tractors or Vice Versa* – The standardization activities of the SAE and ASAE have established dimensional relationships whereby this is successfully accomplished with drawn implements using either a tractor drawbar, a tractor power take-off, or the double-acting remote cylinder of the tractor hydraulic system. It appears also desirable to secure the same provisions for integral implements using a common hitch. To accomplish this desirable result, the industry should make available the following information:

1. Dimensional relationship of points of implement attachment.
2. Arc of travel provided at points of implement attachment by the tractor hydraulic system.
3. Precautions that implement manufacturers must take to secure proper functioning of depth-control means provided in the hitch.
4. Permissible loads that can be safely applied to the hitch.

The 3-point hitch which uses a crossbar length of 26⅞ in. is satisfactory for a 2-plow size tractor; however, this is not the most satisfactory length for a plow in the 3-bottom size. The center of draft of a 2-bottom plow, of 14-in. width per bottom, is approximately 17½ in. from the furrow wall. Allowing for the tilt of the tractor when operating with the right wheel in a furrow, a tractor equipped with 10-in. rear tires set at 56-in. tread permits the draft links to be centered on the tractor and provides satisfactory clearance between the front rolling coulter and the right draft link on turns. The center of plow draft is approximately 1¾ in. to the right of the center of the draft links, which is not considered objectionable. This type of hitch hookup is shown in Fig. 2. A 3-bottom plow, 16-in. width per bottom, equipped with a 26⅞-in. crossbar, is shown in Fig. 3. Here the tractor wheel tread is increased to 60 in. to retain the draft links centered on the tractor in the working position and also retain working clearance for the front rolling coulter. The center of draft of a 3-bottom 16-in. plow is approximately 28 in. from the furrow wall. This results in the line of draft of the plow being approximately 7 in. to the left of the center of the draft links and sets up a horizontal component which produces side draft on the tractor and results in difficult steering.

Our experience has been that a crossbar 32½ in. long, rather than 26⅞ in. as shown by Fig. 4, permits attaching a 3-bottom 16-in. plow in a more favorable position behind a 3-plow size tractor so that the center of draft of the plow can be more nearly aligned on the center of the tractor draft links, thereby reducing side draft on the tractor. Having a center of plow draft slightly to the left of the tractor draft links is often a desirable feature as it compensates for the natural tendency of the tractor to run to the right because of the tractor weight transfer and caster action of the front wheels resulting from the right side of the tractor being low. In addition, the 32½-in. crossbar has several other advantages over the 26⅞-in. bar on the larger implements. It permits attaching a moldboard plow close to the tractor, provides clearance for the front rolling coulter, permits plowing at a wider tractor wheel tread setting, which in turn improves the tractor lateral stability on hillsides. The long crossbar is favorable to rigid

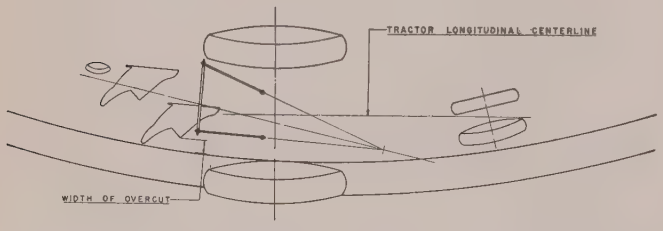

Fig. 5 – Trailing characteristic of a plow on left-hand turn having the hitch point in the horizontal plane located well ahead of tractor rear axle

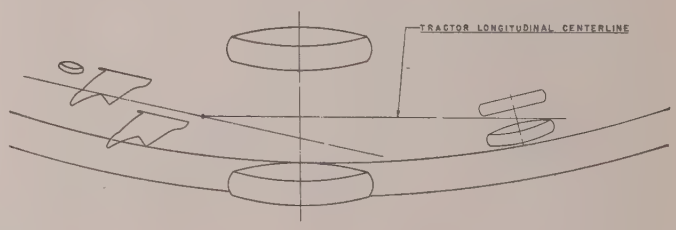

Fig. 6 – Trailing characteristic of a plow on left-hand turn having hitch point in horizontal plane located behind tractor rear axle

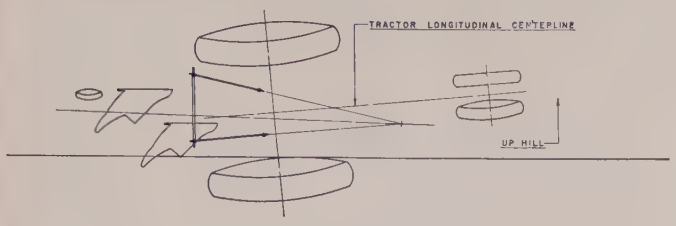

Fig. 7 – Trailing characteristic of plow on hillside having hitch point in horizontal plane located ahead of tractor rear axle

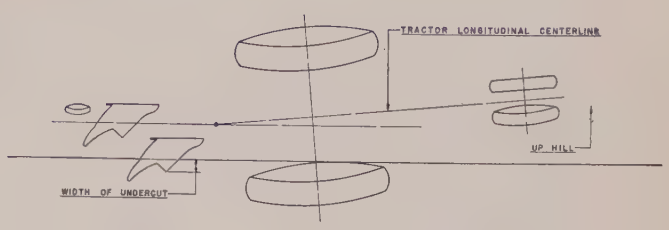

Fig. 8 – Trailing characteristic of plow on hillside having hitch point in horizontal plane located behind tractor rear axle

implement frame construction and implement lateral stability.

B. *Quick and Easy Attachment of Implements* – The integral implements on the market today are generally more difficult to attach or detach than corresponding drawn tools. A design which enables one man to attach an integral implement to a tractor quickly and easily and without undue physical effort is certainly a necessity. Some tractor-hitch designs are such that the tractor driver can attach an implement while sitting in the tractor seat. Such an arrangement is popular with the trade, especially where tractor design is such that it is rather difficult for the operator to climb off and on the seat. A 3-point hitch, as it is known today, is quite successfully connected by one man in instances where the implement can be shifted about by hand to make the necessary alignment for attaching the links. Attaching the heavier implements to such a hitch virtually becomes a two-man job. Quick attachable features should be incorporated in the hitch wherever possible.

C. *Simplicity and Ease of Adjustments*–In order to secure optimum performance from both the tractor and the integral implement under varying field and soil conditions, it is necessary that the hitch be in proper adjustment. The adjusting members should be readily accessible and easy to adjust without the aid of special wrenches. They should also be so positioned that the amount and direction of adjustment necessary is readily detected. To help reduce operator confusion, the number of adjusting members should be held to a minimum. All too often a hitch in improper adjustment results in poor implement performance or heavy implement draft.

D. *Provision for Fore-and-Aft Leveling of the Implement* – Most tillage tools such as the moldboard plow, disc plow, and bedder require a relatively close relationship between the angle of the implement and the soil. This relationship is normally established by adjusting the length of the top link, which on some hitches is also a means of depth control. Fore-and-aft leveling is required to take care of manufacturing variation in implement and tractor parts, variation in rolling radius of pneumatic tires, as well as variations in size and type (cane and rice, and so forth) of tractor tires. Furthermore, a readily adjustable upper link aids in the ease of attaching an implement to a tractor.

E. *Provisions for Lateral Leveling of the Implement* – Provision for lateral adjustment of the implement is necessary to take care of manufacturing variations the same as the fore-and-aft leveling discussed above, and in addition, to provide for the condition where one wheel of the tractor must operate in a furrow for some implements and on level ground for others.

F. *Provision for Side Sway* – To secure the desired trailing characteristic of tillage tools such as plows and tillers, it is necessary that the tool follow the tractor around contours cutting a relatively uniform width without interfering adversely with the tractor steering. Fig. 5 shows a plow having the virtual hitch point in the horizontal plane located well ahead of the tractor rear axle and on the centerline of the tractor when moving in a straight forward direction. On a left-hand turn, the plow will overcut, and the line from the virtual center of forces of the draft links to the point of resistance of the plow passes to the right of the center of the tractor at the rear axle. It can be seen from this figure that the most favorable performance from this standpoint is secured with the point of convergence of the draft links positioned relatively close to the rear axle of the tractor.

Fig. 6 shows the hitch point of the plow in the horizontal plane located behind the tractor rear-axle centerline. Here the plow follows the tractor very well on contours, as the point so located swings in a direction opposite to that which the tractor is turned and thereby aids in maintaining a more nearly uniform width of cut.

In side-hill plowing, the front wheels are usually uphill from their normal position with respect to the furrow, to counteract the tendency of the tractor to drift downhill. Therefore, if the virtual hitch point of the plow is ahead of the rear-axle centerline, it will lead the plow uphill to counteract the tendency to undercut or overcut. With the hitch

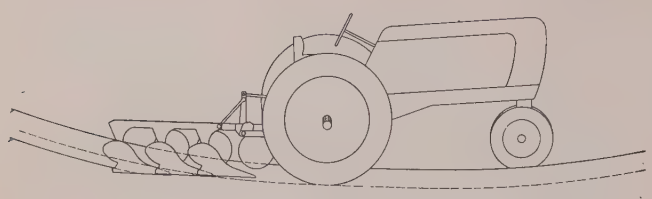

Fig. 9 – Suspended implement operating through a swale

point to the rear of the rear-axle centerline, the plow is actually lead in the wrong direction, causing the plow to overcut or undercut adversely. Figs. 7 and 8 show the effect which the hitch point of the plow has on the trailing characteristics on hillsides. The ease of steering when plowing on a hillside is affected in the same way as it is on contours. When plowing on a hillside, turning furrows downhill, the tractor and plow will be in the same relationship to each other as if they were plowing on a left-hand contour. The moment arm caused by the off-center line of draft makes it difficult to hold the front wheels up the hill or out of the furrow. From this standpoint, a carrier with a large line-of-draft offset will be the most difficult to steer on hillsides and contours.

It appears, therefore, that the best position for the hitch point is near the rear axle and on the tractor centerline, as this will be a compromise between hillside and contour plowing from the standpoint of controlling width of cut; and also, there will be no effect on steering from either an offset line of draft or from the tendency of the plow to rotate in a horizontal plane.

G. *Provisions for Elimination of Side Sway* – It is necessary that implements such as bedders, planters, some types of cultivators, scoops, and the like be locked to the tractor to prevent sidewise movement with respect to the tractor. Therefore, provision should be made to lock the side-sway linkage where required. Furthermore, the implement is transported more satisfactorily if the sway is restricted.

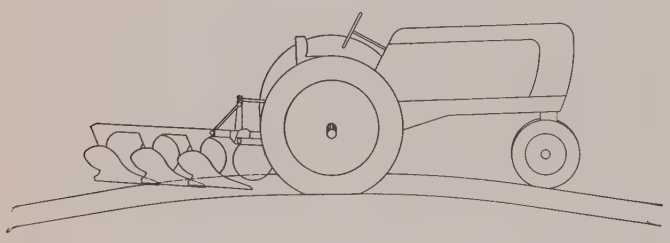

Fig. 10 – Suspended implement operating over hill

H. *Lifting Capacity of Hydraulic System Must be Adequate* – The lifting capacity of the hydraulic system incorporated in most tractors has been developed over the years to meet the requirements of implements in general use. Formerly, however, the hydraulic system of the tractor was used only in raising and lowering implements and not to adjust their working depth. When the hydraulic system is used for depth adjustment as well, capacity must be added for this purpose. In general, approximately 30% of the hydraulic piston travel must be allocated to depth adjustment, and as much of the lifting capacity as was previously required for lifting must be continued for that purpose.

As tractors of higher horsepower are gaining in popularity, so too will larger implements be required to match the draft capacity of the tractor. These larger integral implements such as moldboard plow in the 3- and 4-bottom size have considerable length, and this large overhang requires a hydraulic system of considerable capacity to raise the implement out of the ground.

I. *Means for Depth Control* – The problems of depth control and depth adjustment and the means for providing these functions are by far the most difficult of the entire hitch program. Some implements such as planters, rotary hoes, and spike-tooth harrows are more or less self-gaging as to depth control and therefore pose no particular problems. Moldboard plows, disc plows, listers, subsoilers, and cultivators require depth-gaging means. Furthermore, the operating characteristics of each of these tools are different, and it is therefore difficult to establish one means of depth control that will apply to all. Five basic methods of depth control are in general used with integral tools.

1. Suspension of the implement: By this method, the implement is lowered to a desired working depth by the hydraulic system and suspended from the tractor in this position. The tractor wheels then become the gaging points. This is the simplest method from the mechanical standpoint but has several disadvantages. As the tractor is driven over uneven ground, the implement's working depth varies. If the tractor is operated through a swale, the implement will have a tendency to go deep; and, conversely, if the tractor goes over a hill, the implement will shallow up or come out of the ground.

It is not easy to control the working depth of a rear-mounted tillage tool over uneven terrain, but with a suspended implement the problem is most difficult. A suspended rear-mounted implement operating through a swale and over a hill is shown in Figs. 9 and 10.

2. Independent depth-gaging means provided on the implement: This method of depth control is identical to that employed successfully on drawn implements. However, when applied to integral implements, the economic advantage and simplicity are sacrificed. The working depth of the implement

must be controlled by changing the gage-wheel setting either by pinning the gage wheel at a selected setting and adjusting the depth through a system of levers, or by adjusting the gaging depth by hydraulic means which requires a two-function hydraulic system which few tractors now possess.

3. Depth gaging by adjustment of suction in the implement: This method of gaging is used on some hitches especially with moldboard plows which readily respond to suction adjustment. Its principal advantage is its mechanical simplicity since it lends itself to any hydraulic system. In addition, the fore-and-aft leveling means of the hitch can also be used as a mean of depth adjustment. Implements so controlled normally vary considerably in working depth when operating over uneven ground or in varying soil conditions. Implements that are relatively short, such as subsoilers and some types of cultivators, do not respond to this method of depth control.

4. Depth gaging by adjustment of hitch point in a vertical plane: In this method of depth control, the hitch point is raised or lowered until the implement is working at a desired depth. Although this method is relatively effective, it is not greatly used because the means of accomplishing this adjustment is usually cumbersome and somewhat complicated. A wide range of vertical adjustment is needed to meet the requirements for all implements. The implement using this system usually varies considerably in working depth due to the varying texture of the soil.

5. Depth control by adjustment by draft or load reaction: In this method of depth control and adjustment, the implement draft or load is used as a means for gaging the working depth of the implement. It operates on the assumption that the working depth is a function of the implement draft. A force proportional to the implement draft acts through the hitch and is connected by a linkage to the control mechanism of the hydraulic system. The hydraulic system then acts to raise or lower the implement as the load increases or decreases. The hydraulic system may be used to control the working depth by exerting an upward force on the lower links, which has the effect of raising the virtual hitch point, or it may also be used to alter the implement suction. In addition, the control linkage may be designed so that the hydraulic system operates to maintain the implement at substantially the exact draft value corresponding to the control lever setting, or it may be designed to compensate only partially for the implement draft. In the single- and 2-bottom plow tractor size, this type of depth control in our opinion has the most appeal. A tractor and implement with this type of control can negotiate a very uneven terrain without objectionable variation in the implement working depth where the soil condition is fairly uniform. The tractor can operate through hard or heavy spots in the field without experiencing either traction or power stalls but at a sacrifice of working depth, thus giving the tractor operator a feeling of high horsepower reserve.

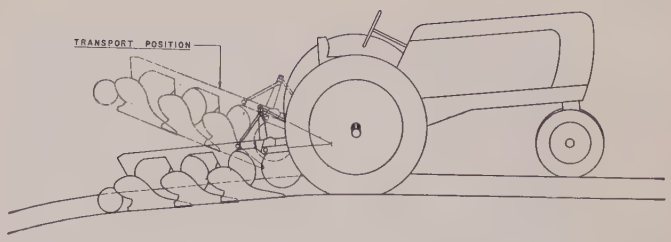

Fig. 11 – Hitch point in vertical plane located near rear-axle shaft

On tractors having a large horsepower reserve for the size implement, a type control which would hold the implement at a constant working depth might be more desirable than load or draft control.

J. *Hitch Should Maintain Implement in Proper Relationship to Ground* – The hitch should be of such design that it will keep the implement in proper relationship to the ground under the implement as the tractor and implement are operated over uneven terrain. It is desirable that the working depth through swales and over hills be about the same as the depth on level ground. The hitch should permit the implement to enter the ground to its working position in a relatively short distance as well as to carry the implement at a safe and secure height above the ground in transport.

A hitch point in the vertical plane located near the tractor rear-axle centerline, as shown by Fig. 11, allows the implement to follow the contour of the terrain through swales and over hills. As the front wheels of the tractor come out of a swale, the attitude of the plow is changed to put more suction in the plow, thereby aiding in retaining plowing depth through a swale. As the front wheels of the tractor pass over a hill, the longitudinal angle of the plow is changed to reduce the suction in the plow and prevent the plow from going too deep. This type of geometry has its objections in that the plow requires longitudinal leveling for various

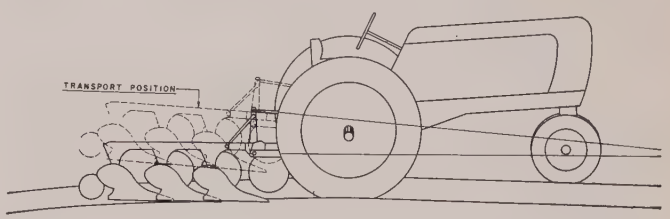

Fig. 12 – Hitch point in vertical plane located well forward of rear-axle shaft

plowing depths because for each change in plowing depth, suction of the plow is also changed. A vertical hitch point so located results in inefficient lifting characteristics and, in addition, also pitches the rear end of the implement to an awkward angle in the raised position. A hitch point in the vertical plane located well forward of the rear axle, as shown in Fig. 12, changes the longitudinal angle of the plow very slightly as the plow moves up or down with respect to the tractor. This action causes a free-floating plow to bridge across swales and plow excessively deep over hills. However, it does permit plowing at various depths without making adjustment in the longitudinal leveling of the plow. Tractors having draft or load control hydraulic systems usually have such a vertical hitch point, but here the plowing depth is controlled by the tractor hydraulic system.

Therefore, it appears that a compromise must be made in the hitch-point position in the vertical plane for free-floating implements so that the implement may be maintained in the most desirable relationship with the ground, both in working and in the transport positions.

K. *Provision for Remote Cylinder* – With the trend toward tractors of increased horsepower and greater versatility and integral implements of larger size, provision should be made in the tractor for a hydraulic remote cylinder which can be operated independently from and simultaneously with the tractor rockshaft. A remote cylinder can be used effectively in operating gage wheels on implements requiring them, reversing two-way tillage tools, and for controlling drawn tools equipped for remote cylinder operation. In addition, remote cylinders are used extensively for operating front-end loaders, scoops, and cranes around the farm buildings.

L. *Provide Satisfactory Tractor Longitudinal Stability* – To transport the larger or heavier rear-mounted integral tools with safety, adequate weight must be retained on the tractor front wheels so that the tractor can be steered safely over rough terrain or up inclines. Recognizing that rear-mounted pickup tools are here to stay, more tractor front-end weight is required than has been necessary in the past. Some means for readily attaching a reasonable amount of ballast to the tractor front end, when heavy rear-mounted tools are used, should be provided, and yet this weight must be readily removable when implements are mounted on the front of the tractor.

Implement Design Features for Satisfactory Operation

There are a number of implement design features that probably will have to be the responsibility of the individual implement designer and will have to be provided as a part of those implements requiring them.

A. *Provision for Landing the Implement* – On hillside or contour operation, it is often desired to maintain the width of cut of the implement. This is accomplished by means of a landing device which adjusts the implement side-wise with respect to the tractor.

B. *Provision for Overload Protective Devices Where Required* – Tillage tools used in rocky or stumpy land and attached to a tractor of high horsepower require some protective device to prevent damage to the implement frame or soil-working elements. This is normally accomplished by means of shear pins or a type of spring release or trip mechanism.

C. *Independent Depth-Gaging Means* – Some implements, because of their size or special operating characteristics, may not respond properly to the depth-control means provided on the tractor. These tools may require independent gage wheels. The depth is varied by means of a lever or screw crank which can be reached from the tractor seat or by use of a remote cylinder which can be operated from the tractor hydraulic system.

D. *Provision for Allowing Implement to Float Laterally* – Implements such as a four-row lister are difficult to control laterally without the use of gage wheels, especially over uneven terrain. If the lister bottoms on the right side are operating at a greater depth than those on the left, the tractor becomes difficult if not impossible to steer. Such implements, therefore, require lateral movement or float between the hitch and the implement. Planters also require lateral float in order that the planting depth can be controlled over rolling ground conditions.

E. *Tractor Longitudinal Stability* – The design of rear-mounted integral implement of a size which equals the drawbar capacity of the tractor often requires that the weight and length of the implement be held to a minimum and that the implement be positioned as closely to the tractor as possible. With the implement in transport position, sufficient weight must be retained on the tractor front wheels so that the tractor with implement can be transported safely in fields and on roadways. If satisfactory tractor stability is not obtained by positioning the implement or by reduction of implement weight, then it will be necessary to resort to adding ballast to the tractor front end.

Conclusion

The different types of tractor hitches for integral implements on the market today have different operating characteristics on a common type implement. A hitch that performs satisfactorily on a moldboard plow may not operate to any degree of satisfaction on a lister, cultivator, or subsoiler. Each of the types of hitches covered in this paper performs better on some tools than on others. A type of a control that will satisfactorily operate a wide variety of tools is still desired.

An implement designer's viewpoint on tractor hitches and hydraulic systems is presented in the hope that tractors in the future will have common implement-to-hitch attaching points so that a tractor may be used with a number of different makes and sizes of integral implements.

DISCUSSION

Design Problems Of Larger Tractors

– W. E. Knapp
Minneapolis-Moline Co.

THE author has very adequately discussed the performance features desired in the design of the hydraulic system and implement hitch. These features are generally a consideration of the tractor engineers. Just how the implement designer uses them is his problem in each individual case. There are a variety of implement hitches on the market today – some permit a great deal of implement interchangeability while some other designs can accommodate many by the use of adapters. Regardless of the type of hitch, all of the various tractor-implement combinations fulfill most of the performance features outlined by Mr. Johannsen. The larger tractors and their implement hitches have not been available to the extent as the 2-plow size. This is significant and indicates that the designer's problem is more complex. Naturally the same performance features are required for the larger tractor hydraulic system and implement hitch. But the implement engineers have not, as yet, been able to suggest to the tractor engineer group just what is desired in the way of implement performance. They, of course, would like performance equal or better than that obtained with the present 2-plow-size implements – but realizing that many additional factors are introduced with larger tractor implements, the designers have not evaluated all the features either from a performance or economic standpoint.

Some factors that must be considered for larger tractors are:

1. The greater power and traction available in these tractors will necessitate heavier hitches.

2. The greater inertia or kinetic energy they possess because of their greater weight and higher speeds will exert far greater shock loads on the hitch and implements. Most implement damage is the result of shock loads, and protective devices will have to be used for many different implements.

3. The hydraulic system will have to be commensurate with the larger equipment and have a lifting capacity probably well over 2500 lb. This will certainly require some additional ballast to the front end of the tractor.

4. The weight of heavier implements and ballast to the larger tractors may require larger or heavier tires to support them.

5. The larger size will probably make getting on and off the tractor more difficult and should make attaching from the tractor seat a more pertinent consideration.

The smaller-size tractors with hydraulic hitches are very often used for work other than farming. Many of these tasks are incidental to farming, such as building and maintaining farm roads, making farm ponds, drilling fence post holes, and numerous other jobs. The larger tractors with hydraulic hitches will do many of these jobs and faster because of increased power. But it is my belief that their principal use will be for farming operations. If this is true, then more consideration should be given to the strictly farm implements and their performance, with almost total disregard to their interchangeability with those implements and attachments now available for the 2-plow-size tractor. The implements most generally used and which must be considered for use with the larger 3- and 4-plow-size tractors fall into two principal groups: those that are quite long and frequently heavy and implements that are wide. The first category will include:

1. Three- and 4-disc plows.
2. Three- and 4-bottom moldboard plows.
3. Tandem disc harrows.
4. One-way disc tillers.

The wide-type combinations will include:

1. Three- and 4-row listers or middlebreakers.
2. Four-row cultivators.
3. Tool-bar field cultivators.
4. Drag harrows.
5. Four-row planters.

In the longer-length group, we find that some require little or no depth control from the tractor. These would be the one-way disc plow and tandem disc harrows. They may, in fact, require additional weight for penetration. Others, such as the moldboard plows and disc plows, will require adequate depth control. The depth control can operate directly through the hitch or actuate gage wheels. The most desirable would operate on a constant depth principle and maintain this depth automatically.

When the tractor is used to control the operating depth of various implements through automatic load or depth control, a great degree of rigidity is required in both hitch and implement. Excessive deflections in either will cause variations in the operating depth. With larger-size units, such as 3- and 4-bottom moldboard plows, the deflections may cause the plow to vary excessively in depth despite the corrections continually being regulated by the hydraulic system. These deflections can be minimized by using much heavier sections for the beams and hitch parts. When much heavier sections are used, naturally a resulting increase in cost is experienced and possibly to such an extent that the price advantage discussed by Mr. Johannsen in the case of a mounted 2-bottom plow is lost on larger-size implements.

If it is necessary to secure adequate control by the addition of gage wheels and accompanying levers or hydraulic control, the cost will certainly increase and may well exceed that of towed-type implements.

Mr. Johannsen has brought out that one principal feature contributing to the popularity of the integral implements is their greater maneuverability, aiding in working small or irregular shaped fields. In general, the larger tractors are used on larger farms containing few small or irregular fields. These larger units are more often equipped for 4-row operation and require ample headlands or turning space. It thus seems that maneuverability may not be too great a consideration with larger tractor implements.

If there is little economy to be gained through a hitch and implements for larger-size tractors and little requirement for maneuverability, one may ask, "Why consider hitches and equipment at all for larger tractors?" Perhaps it would be sufficient to equip these tractors with hitches identical to those used on the 2-plow-size tractor. They could then perform all the jobs incidental to farming and, by using conventional mounted or towed-type implements, handle the farming operations. Despite these considerations, the farmers are demanding more mounted equipment of all sizes in preference to towed type. It is only natural that a single hitch should be employed to attach all of these to the tractor.

Frequently there are many implements used in com-

binations, particularly on the larger tractors. Some of these may be a cultivator mounted on the front of the tractor and a planter on the rear, or a fertilizer attachment on front and a planter to the rear. Even row crop cultivators are generally front mounted with shovels or sweeps to the rear to remove wheel tracks. Naturally, the rear portions of such combinations would be mounted on the hydraulic hitch when the tractor is so equipped. These considerations further complicate the implement designer's problem. I have limited this discussion, so far, to tillage and allied implements. Perhaps we should ask, "What is the ideal hitch?" "What should be expected of such a hitch?" There are possibilities of using the hitch for mounting implements other than tillage. "Would it be desirable and economical to mount portions of a corn picker on this hitch?" "How about a baler or other heavy harvesting equipment?" These are possibly a little far-fetched but I feel certainly must be given every consideration.

Mr. Johannsen has given an excellent presentation on the subject of hitches. It has been impossible to find something to discuss which he has not covered. The ideal hitch, in my opinion, is not here today, but from the many approaches which all companies are taking, it perhaps will not be long before it will evolve. The implement engineers cannot accomplish this alone – it will require a very closely coordinated development program with the tractor engineers and eventually an attempt at an industry-wide program to obtain interchangeability of various makes and models of both tractors and implements.

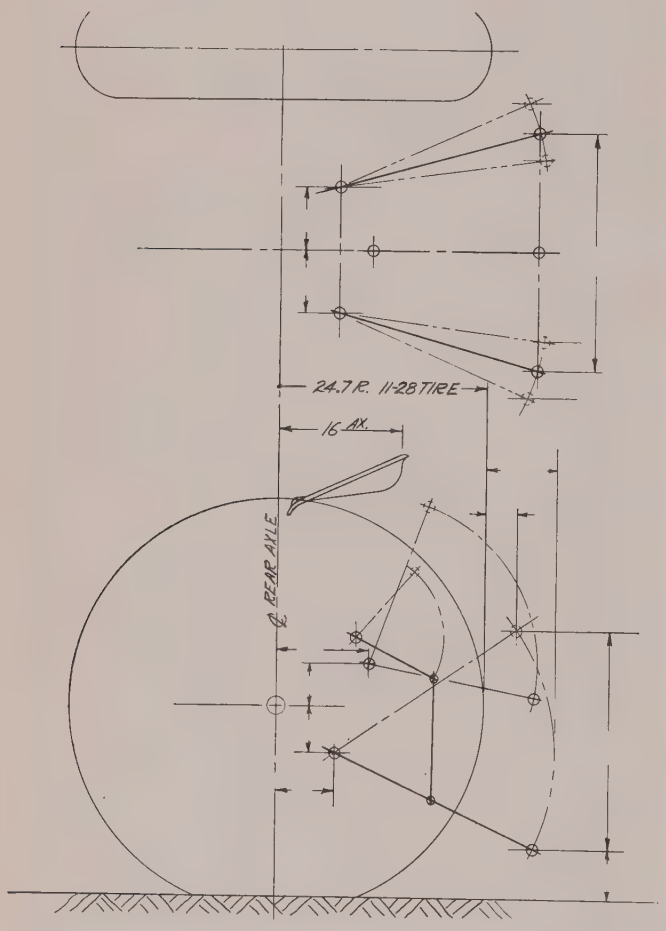

Fig. A – Relationship of hitch points to ground, to tires, and to seat

Emphasizes Need for Hitch-Point Standard

– E. W. Todd
Massey-Harris Co.

THE advantages of integral-mounted implements are listed as greater maneuverability for working in small or irregular shaped fields, better and faster transportation over hard-surfaced roads, lower cost to the farmer. These advantages are becoming more pronounced for the reasons stated.

Convenience features desired are listed as:

1. Provision for implement to be used with competitive tractors.
2. Quick and easy attachment of implement.
3. Simplicity and ease of adjustment.

The provision for the implement to be used with competitive tractors is desirable from the customer's standpoint and therefore should probably have consideration by this industry. Mr. Johannsen lists four items of information that should be made available by the tractor manufacturers to make this interchangeability possible:

1. Dimensional relationship of points of implement attachment.
2. Arc of travel provided at points of implement attachment by tractor hydraulic system.
3. Precautions that implement manufacturers must take to secure proper functioning of depth-control means provided in the hitch.
4. Permissible loads that can be safely applied to the hitch.

A diagram as shown in Fig. A could be used to give the dimensional relationship of the points of implement attachment to the tractor. This diagram, you will notice, gives the relationship of the hitch points to the ground, to the tires, and to the seat. Other pertinent information could be supplied as required.

We might add here that if there were some standardization of the height of the hitch points on the implement (taking into consideration the depth at which the implement normally works), it would simplify the hitch designer's problems.

In regard to two different lengths of crossbars being desirable for 2- and 3-bottom plows, we are in full agreement with the thinking that a crossbar longer than 26⅞ in. is desirable for 3-bottom plows and other implements used on larger-size tractors.

It might be well to keep in mind that there is a proposed British standard for 3-point hitches covering points of implement attachment for the 26⅞-in. crossbar and for a wider crossbar which they refer to as the No. 1 and No. 2 proposed standard. Some of the dimensions are shown in Fig. B. If the proposed British standards for 3-point hitches are generally acceptable and followed, it will facilitate interchangeability of implements in the export field.

Quick and easy attachment of implements is desirable. It is essential that one man be able to hitch a tractor to a heavy implement quickly and with a minimum of physical effort. It is not easy for the average tractor operator to align the tractor exactly with the implement so it can be attached easily regardless of whether it is latched or pinned into position. Jostling of the implement by hand or with the tractor on the concrete floor of a machine shed to get it in alignment should not be necessary, and if it is done, it may damage some parts of the machine that are in contact with the floor. It seems that a hitch

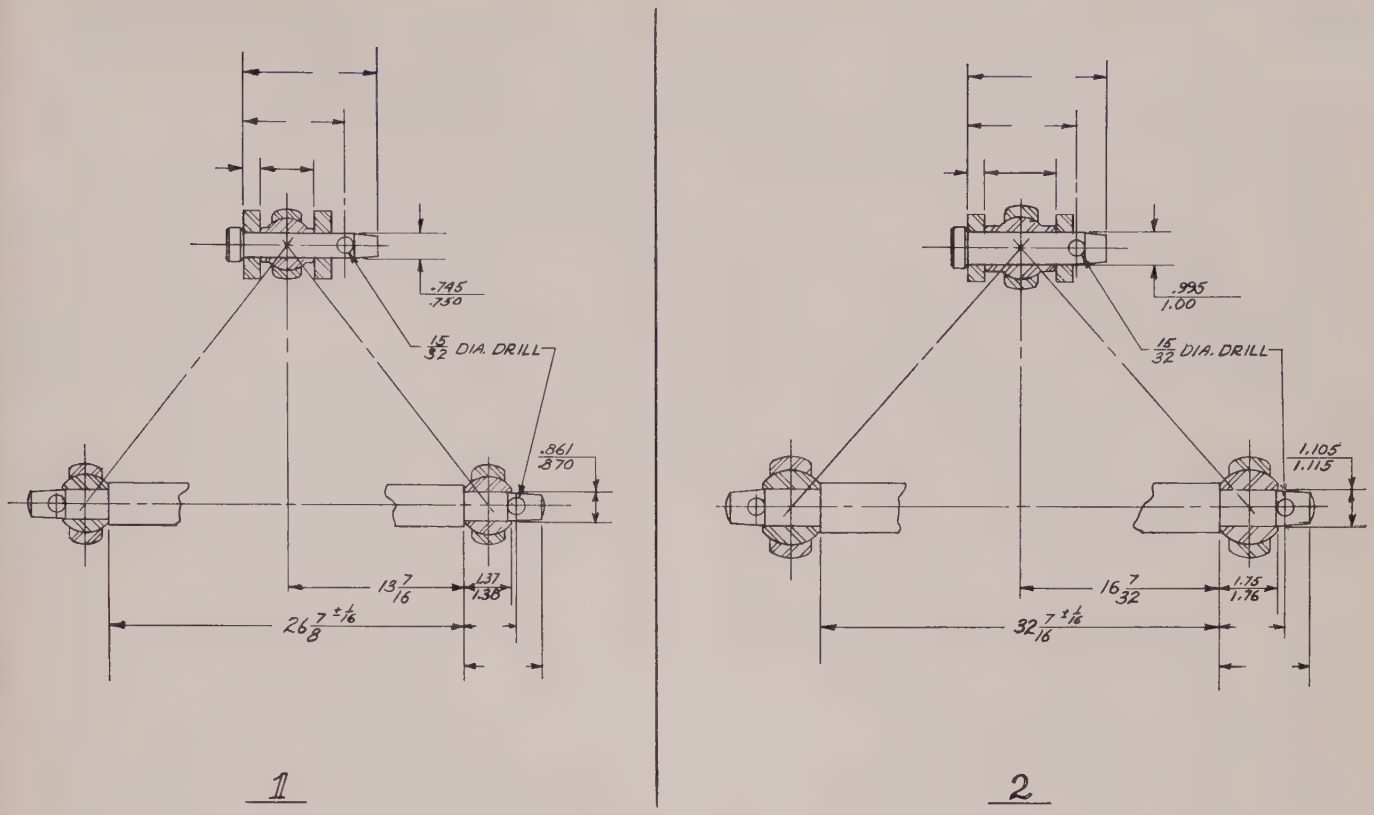

Fig. B – Some dimensions of proposed British standard for 3-point hitches

could be provided so that the tractor does not have to be in exact alignment for attaching.

The need for simplicity and ease of adjustments has been very well covered. We might add that any adjustments that require frequent attention should be adjustable from the tractor seat.

The advantage of integral-mounted implements and the necessity for a quick and easy attachment of implements has been discussed. The most important phase, however, is the performance of the tractor and implement combination. Many factors affect performance. The principal factors being:

1. The geometry of the hitch.
2. Suitable means of depth control.
3. Provision for the implement to trail on contours and provision to lock out trailing feature when desirable.
4. Provision for independent vertical flexibility of the hitch members and provision for locking the independent vertical flexibility when desirable.

The geometry of the hitch affects the ability of the implement to enter and stay in the ground, the ability of the plow to work at a fairly uniform depth over uneven ground, the amount of weight transferred to the tractor, the width of cut when plowing on contours, and other items of less importance. Incidentally the practice of listing several different diameter tires for a tractor does not contribute to good performance as raising or lowering of the tractor affects the relationship of the hitch and implement combination to the soil. This practice should be avoided.

Five methods of depth control are listed by Mr. Johannsen. Without going into considerable detail, there is very little comment to be made on this except to say that if the hydraulic system and hitch provides for more than one means of depth control, better performance will be obtained from a wider range of implements.

The lifting capacity of the hydraulic system should be at least 25% greater than that required to lift the maximum weight recommended for the tractor in transport. The reserve is necessary to lift the implement out of the ground and to provide for momentary loads.

In concluding this discussion, I want to say that Mr. Johannsen has outlined the implement designer's problems in connection with the hydraulic systems and hitches very well. He has suggested that the implement attaching points on the tractor hitches be standardized to permit interchangeability of competitive implements. This suggestion should have further consideration.

Longer Crossbar May Reduce Interchangeability

– L. G. Kopp
Oliver Corp.

WE concur with the points that fully mounted implements permit better transport conditions and also result in substantial lower cost per implement.

It has been our experience that any drawn implement which has an automatic coupling point and a self-supporting hitch may be coupled more rapidly than any other multiple attaching point mounted implement. Unfortunately, many of the drawn implements do not have both characteristics of an automatic coupling point and self-support hitch.

For instance, many drawn implements do not have an automatic coupling point when a simple hitch pin is provided. Many implements with multiple attaching points have been well accepted by the public, which are not as fast as the single attaching point.

We believe that the sacrifice of adjustment or some other feature to gain the simplicity of an integral-mounted implement has been a desirable choice. In many cases, a customer user is unwilling to use an implement with miscellaneous adjustments. A few areas of our country have employed operators of farm equipment who do not use implement adjustments successfully.

Mr. Johannsen, in reviewing the possibilities of standardization of swinging link hitches, mentioned the desirability of increasing the length of the attaching crossbar. We have observed such a desirability; however, we felt that it tends to reduce general interchangeability of various sizes of implements between different sizes of tractors. This feature has been acceptable at the 26⅞-in. crossbar length so long as tractor power and weight did not exceed the 3-bottom plow classification in size. Other considerations affecting the choice of implement crossbar length are the ease of adjusting and range of tractor rear-wheel tread; some tractor operators are unwilling to adjust wheel treads which, in turn, may cause off-center loading and result in difficult steering.

It is desirable to hold the intersection of the mounted implement swinging links near the rear axle as Mr. Johannsen mentions, except that we believe it is a penalty to have this intersection point so near the rear axle that wide tool bars must always be restricted against sideward movement with respect to the tractor.

A review of the remarks about hydraulic system lifting capacity being adequate would imply that this might frequently be a problem. Our observations are that lifting capacity within the hydraulic system is quite readily secured. This in turn, creates a much more serious problem, which is adequate transport stability. Transport stability also is much more of a problem as tractor power and implement center of gravity relationship get farther and farther away from a 2-plow-size tractor and implement combination. We believe that this factor alone is the main point which will cause difficulty in providing satisfactory mounted implements for the 4-bottom plow and larger-size tractors. Semimounted implements have never been very popular in the past. With improved attaching points, semimounted implements can become a solution for tractor transport stability with large tractors of the present design arrangement.

We believe that the objective of having common implement-to-tractor attaching points is good. We, however, anticipate that the constant search for improved products and functional characteristics will cause a constant change, such that a satisfactory tractor-to-mounted-implement hitch standard will not be established or remain in effect for any reasonable period of time.

Other Disadvantages of Integral Implements

– S. C. Heth
J. I. Case Co.

THERE is no doubt but what the advantages of mounting many implements will expand the market for tractors equipped easily to pickup, carry, and operate such tools – this in spite of the disadvantages of mounting some tools, which are not negligible. The major disadvantages, which implement men should not lose sight of, are:

1. Weight required for penetration, when needed "way back," is difficult to apply because of moment-arm lengths and necessity of maintaining some weight on the steering wheels when implement is lifted. This, of course, applies to contemporary, rear-positioned mountings.

2. Down-pressured carrier arms, to "add weight" to implements, "removes weight" from tractor rear wheels with obvious ill results.

3. Resultant of forces in larger implements, particularly plows, are not constant for all sizes, number of bottoms, and so forth; and projected draft lines will vary. Good operation will necessitate specific implements for specific tractors, or at least specific ranges of tractors, and reduce interchangeable possibilities, regardless of even optimum standardization, which is still in the far distance.

4. The matter of quick attaching is open to question. Actually, it is my opinion that most mounted implements are slower to hook up than most pull-type implements, under the conditions actually encountered on the farm. There are advantages in transport; there are some advantages in operation; in some cases, hookup is easier in terms of manual effort. In terms of quick hookup, engineers haven't improved the pull-type situation; quite the contrary is correct, particularly with smaller tractors and implements. The hookup advantages become more apparent in larger tools and tractors.

5. There is necessity of special pivoting arrangements and the like on some tools, particularly plows, when operated on contour. This is also a factor in noncontour farming, as mounted tools without pivoting arrangements will do much to "steer tractors," possibly more than the wheels provided for that purpose.

6. Mr. Johannsen says that planters and some other tools are more or less self-gaging as to depth control. It is my opinion that most are mostly less self-gaging, unless provided with carrier or gage wheels.

I don't agree on need for a 32½-in. crossbar. Argument for more clearance on some tools is legitimate, but I don't follow reasoning that it creates a better situation in regard to draft lines. Why not move the crossbar on hitch pins on the implement to the left to obtain correct lines of draft?

In regard to discussion of projected hitch point, there are times when it is desirable to have it considerably forward of the rear axle, in fact, in front of the front axle. Rear-mounted cultivators require the projected hitch point forward in order to work properly. Of course, the whole question of rear-mounted cultivators can be debated.

While standards of some nature are essential and will come about eventually, the problem is not simple. Among the situations engineers will face are the hitch-point-arc and power take-off relationships, and the hitch point to tire relationships. The arc of power lift-hitch combinations, for rear of tractors, is so new that some time will necessarily lapse before developments reach a stage where our customers will, by their selections, force us all to acknowledge the best relationships and design.

Type of Tractor is Important Consideration

– E. W. Tanquary
International Harvester Co.

WE have in this paper an excellent analysis of tractor hitches and hydraulic systems from the standpoint of an implement engineer. Frankly, it contains reference material of value to any engineer interested in direct-mounted implement and hitch design.

Selection of a combination hydraulic system and implement hitch involves many factors. Perhaps the most important is to provide the versatility required for the full range of rear-mounted implements – moldboard and disc plows, mowers, middlebusters, and many other tools. Our experience has been that the farmer is willing to accept a certain amount of compromise to obtain the benefit of quick attachability, but in the end he will insist on field performance or the ability to do the job he wants done.

In addition to providing for rear-mounted implements, it is important that the tractor can be used for regular drawbar work and for power take-off operated implements

with a minimum amount of parts and in a minimum time. In other words, the time required to change over from mounted implements to drawbar or power take-off operation is just as much a part of the attachability problem as the time involved in changing from one mounted implement to another.

The requirements for various tools have been covered in the original paper, but it will be well to add that where depth control is obtained by adjustment of the hitch point it is important to have some means for locking the hitch for use with mowers and power take-off implements.

Provision is also needed for allowing the rear of the hitch members to float when used with planters, tool bars, and similar implements equipped with their own gaging means. Provision for maintaining these members in a fixed, horizontal plane for power take-off is also necessary.

The type of tractor for which the hitch is intended is also an important factor: for instance, the utility-type tractor, with smaller tires and limited clearance under the main frame and clutch housing, does not lend itself to depth control by adjustment of the hitch point. Here the 3-point linkage seems to be the best solution, provided the rear of the tractor permits locating the implement close up to the tractor without resorting to extremely short linkage.

On other tractors, with ample clearance under the main frame and clutch housing but with the belt pulley and power take-off and other units extending pretty far back on the tractor, the 3-point linkage does not work out too well, because of the necessity of using extremely short linkage to avoid setting the implement back so far that the front-end weight of the tractor is affected.

For the tricycle-type tractor the hydraulic system should be designed to provide means for independent adjustment of the front units of front- or side-mounted planters and cultivators. The hitch should be designed to retain the benefit of the front-mounted equipment, as well as making provision for the rear-mounted implements.

Many of our customers still like to have the planter units and cultivating equipment in front where they can keep an eye on the performance in the field.

A further consideration is the means for depth control. The relative merits of depth control by adjustment or load reaction as compared to some of the other methods has previously been discussed.

We have chosen to retain the control of depth by adjustment of the hitch point in a vertical plane. We feel that by this method the operator can maintain a more even operating depth under varying conditions and that the disadvantages can be overcome by coordinating the hitch design with the tractor and hydraulic system so that the unit can be operated and adjusted conveniently by the operator.

The means for controlling depth will govern some of the other provisions in the hitch such as we have outlined and, to some extent, design of the implement.

We see no objection to setting up a recommended practice covering the three attaching points for implements designed for use with the 3-point linkage as a guide for small manufacturers in developing local tools to work with these units.

We would, however, like to point out that while establishing these common points will permit mechanical interchangeability, it will not insure interchangeability from a functional standpoint: for instance, moldboard plows designed for use with tractors equipped with draft control are usually built with increased suction in the bottoms to assist in obtaining penetration under adverse conditions.

When used with hitches designed for another type of depth control, penetration is generally found to be excessive, necessitating the use of gage wheels to prevent the plow from digging in when operating in soft ground.

The draft-control or three-point-type hitch has a converging linkage; consequently any vertical movement of the plow unit tends to throw the plow out of level horizontally. A conventional plow without the additional bottom suck, when assuming a lower setting would bear on the heel of the landside and theoretically gage its own depth in that direction. It would also be impossible to adjust for deeper plowing without repositioning the upper link.

In conclusion, I would like to concur with Mr. Kopp in that with the constant search for improved products the method of attaching direct-connected implements, whether front, side mounted, or rear mounted, will undergo a constant change. Two manufacturers already have automatic coupling devices for rear-mounted implements, and there will be still further improvement.

It is our job to see that they are properly designed, that we maintain the proper stability, that the tractor and hydraulic system are not overloaded, and that the complete unit will permit the farmer to perform satisfactorily the necessary field operation, as well as to provide quick and easy attachability.

ORAL DISCUSSION

Reported by C. L. Zink
Deere & Co.

Mr. Johannsen: Mr. Heth does not agree on the need for a 32½-in. crossbar and does not follow the reasoning that it creates a better situation in regards to draft lines and asks, why not move the crossbar on hitch pins on the implement to the left to obtain correct line of draft?

We find that the longer crossbar (32½-in. long) permits placing the line of draft of a 3-bottom moldboard plow closer to the centerline of the tractor than is possible with the 26⅞-in. crossbar, thereby reducing the side draft on the tractor. The longer crossbar also permits retaining the plow close to the tractor for good performance and at the same time provides clearance for the front rolling coulter on right-hand turns. Moving the hitch points to the left on the implement would likely result in interference between the coulter and the right draft link on right-hand turns.

E. W. Tanquary, International Harvester Co.: Here are a few words on the requirement for and the effectiveness of the double-action feature of the International Harvester Super C hydraulic system. Double action was first used for the remote cylinder control of trailing implements such as disc harrows and plows. On the Super C, the remote cylinder is used to raise and lower rear-mounted implements. The double-action feature may be used to provide down pressure on rear cultivator rigs, disc plows, or other implements, or at the option of the operator the tools may be allowed to float.

C. N. Hinkle, Standard Oil Co.: I want to urge tractor and implement design engineers to use the most common hydraulic media – good motor oil. This will normally have had added to it an anti-foaming ingredient, and it should not be necessary to make such extreme demands upon a hydraulic system that a special fluid would be required.

C. A. Hubert, International Harvester Co.: Make the whole hydraulic system as flexible or versatile as possible with respect both to various tractor models and to implement applications. I'd also like to suggest that cylinder size, pump capacity, and other similar components be subject to modification so as to meet new requirements or better handle the old ones.

Tractor Hitches and Hydraulic Systems—

TRACTORS of the future will reflect the integration of knowledge and experience of both the tractor and implement engineer. During the past 15 years a definite trend has been established to design the tractor and its implements as a unit. This trend has resulted from the need of the farmer for a more compact, efficient, and specialized medium to provide less costly production. Present-day farm management has become as cost conscious as our industrial managers, therefore their investments are being channeled into a program of obtaining timesaving devices to combat the continuously rising labor costs.

The part that hydraulics plays in the modern farm tractor to obtain these new goals of achievement is as important a factor as it is in the industrial machinery that equips our modern factories. Belt-driven, mechanically controlled, metal-working equipment, and tractor-drawn implements mechanically controlled are rapidly becoming collectors' items.

With the advent of the tractor and integral implements, rapid strides have been made to develop controls that are both efficient in operation and trouble-free in service. The early mechanical devices soon gave way to the more effective hydraulic components. Impetus was given to the adoption of hydraulics in tractors as a result of Ford offering a tractor in 1939, which embodied a 3-point hitch and hydraulic controls as standard equipment. By so doing, a pattern was established that has resulted in many competitive tractors now being offered with similar features.

Inasmuch as competition serves as the springboard to more advanced design, a continuous and accelerated program is being carried on by the industry as a whole, to provide better hydraulic controls and hitches in their new models of tractors. No other part of the tractor has received so much attention in recent years. The inherent problems of designing the tractor and implement as a package have been recognized and met by many manufacturers by combining the engineering groups into one well-knit organization. This seems to indicate that while implements can be controlled by remote methods, they cannot be designed in a like manner.

The purpose of this paper is to present the writer's viewpoint on some of the many tractor design considerations that must be recognized when incorporating one of the three following types of hitches: the front and rear integral hitch, the towed implement hitch, and the rear integral hitch. Because the tractor's size and configuration are determined largely by the position and methods of hitching and controlling implements, this paper will attempt to point out the effects upon the vehicle as a result of these factors. In view of the fact that there are many types of hitches and hydraulic systems on the various makes of tractors today, it is hoped that this presentation will help to stimulate both prepared and open discussion of the relative merits of the different type units. Inasmuch as the subject matter is rather controversial and the writer realizes no protection from the Fifth Amendment to the Constitution, it is hoped that the deficiencies of this paper will be far overshadowed by the excellent addition of prepared discussion.

The subject is so broad in aspect that the paper can only deal in generalities instead of covering the detail of construction of the various designs. In order to accomplish a comparison of the three types of hitches, this paper will devote one section to each of the designs.

Front-and-Rear-Mounted versus Rear-Mounted Hitch

One of the most controversial phases of tractor operation is that of determining the relative merits of front-mounted cultivators versus rear. It is recognized that there are proponents of both methods of cultivation, just as there are coffee and tea drinkers, throughout the rural sections. However, this dissimilarity of opinion is not a barometer that one design is better than the other but only indicates that farmers are independent thinkers.

While the writer believes there is little justification for a tricycle tractor, the answer lies in the fact that many of these models are sold every year, therefore it would appear that the design is justified. However, it would appear that the four-wheel adjustable-front-tread model of tractor is advancing in popularity and has been well accepted in the corn-belt areas. While front cultivators have been applied to tractors such as Ford, the applications have not been too satisfactory. Because the more successful tractors are built around the front hitch and implements we should examine carefully the effect upon the chassis design as a result of providing clearance for the front tool bars and implements. As a means of accomplishing this, the

A Tractor Designer's Viewpoint

Harold L. Brock, Ford Motor Co.

This paper was presented at the SAE National Tractor Meeting, Milwaukee, Sept. 17, 1953.

comparisons drawn in this section of the paper are relative to a tractor such as we produce.

It is inherent in tricycle tractor design that sufficient undercarriage clearance be provided to permit adequate stroke of the front-mounted tools, therefore it becomes necessary to raise the front of the chassis as much as 10 in. In order to regain the longitudinal stability lost as a result of raising the center of gravity of the tractor, it then becomes necessary to increase the wheelbase or mount heavy weights on the front when operating with rear equipment. The lateral stability of the tractor is seriously decreased both by the higher center of gravity and the reduced horizontal distance between this center of mass and the ground contact line connecting the points of support of the outer wheels. Furthermore, the trend toward increased power-to-weight ratios and faster ground speeds has accentuated the designers' problems in providing a vehicle offering maximum safety without penalizing the operational characteristics.

The increased height of the tricycle tractor presents additional problems in providing easy access to the driver's seat, as well as in furnishing adequate seating comfort. Careful analysis of these two factors indicates a need for further design study to minimize operator fatigue. It is obvious that for every inch the seat is raised above the ground line, a proportional increase is brought about in the lateral and longitudinal motion of the operator when operating over rough terrain. As a result of this condition the problem of seating requires greater attention. No less important is the fact that with front-mounted equipment it often requires a contortionist to climb over the necessary linkage or hydraulic lines provided to control both sections of the implement. Because certain operations require the operator to mount and demount many times to make adjustments or to remove trash and obstructions from the implement, careful consideration must be given to the problem of mounting the tractor.

Now that consideration has been given to the effects of raising the tractor, it would be well to examine some of the operational problems of the pedestal-type front structure as a result of applying the front hitch. In order to provide sufficient clearance for implements the wheels must be tucked within the confines of the chassis, thereby making it impossible, in most cases, for the operator to see what he is running over. This often results in serious damage to the front mechanism because of its characteristic inability to absorb high inertia loadings. A hinged axle equalizes the loading on the two wheels and halves the shock on the radiator and front structure.

Consideration must also be given to the problem of providing adequate front-end flotation when heavy equipment is carried on the front hitch. The tractor designer's problem is magnified because of space limitations as well as that of supporting the tractor within a relatively small area of ground contact. As a result, the possibilities of bogging down and locking the wheels due to soil pickup are much greater than with a hinged axle. When the wheels are spread apart it seldom happens that both wheels become immobilized at the same time. In addition to the problem of flotation, the designer must pay a great deal more attention to the steering-gear design on a tricycle tractor because of the incidence of shocks being imposed upon one wheel without the benefit of damping action by the other member.

We have examined some of the problems of the tractor chassis engineer in providing a suitable vehicle for mounting a front hitch, therefore it would be well to determine how such a hitch affects the implement controls engineer. His most apparent problem to overcome is that of providing the controls and hitch for the front unit at no extra cost, inasmuch as a rear unit must be provided, as would be the case of a tractor embodying only a rear hitch. Because dual hitches require dual controls, it then becomes necessary to provide delayed-action mechanical or hydraulic mechanisms in order to meet certain operational requirements.

Before the implement engineer begins to feel

The Author

HAROLD L. BROCK (M '44) has been chief tractor engineer of the Ford Tractor Division, Ford Motor Co. since 1940. He joined Ford in 1930 after graduating from the Ford Trade and Engineering School and attending the Detroit Institute of Technology. During World War II, Mr. Brock assisted in the design and development of the M4 Sherman tank at the Aberdeen Proving Grounds.

THIS paper presents one writer's viewpoint on some of the many tractor design considerations that must be recognized when incorporating one of the following types of hitches: Front and rear integral hitch, towed implement hitch, and rear integral hitch.

concerned about the tractor engineer's problems of adopting a front hitch to the vehicle, it would be well to examine some of his problems as a result of making the hitch available. To take advantage of the hitch, the implement engineer designs many of the implements so that portions are mounted on the front as well as the rear. By following this line of thinking, the implement engineer, in most cases, has doubled everyone's problems. He must design a double structure for his soil-working tools; the tractor engineer must design dual controls; the manufacturer must double his tooling; and the sales people must convince the public the combination is worth twice as much. The complete package becomes more costly to the operator, both in initial investment and in time lost collecting the parts and assembling them onto the tractor each time they are required.

In many cases, an operator, having a tractor equipped with a complete implement mounted on the rear hitch, can complete the operation during the gathering-together-and-hitching-up period of a front-and-rear-mounted type of implement. Because the front-and-rear implement structures are separated, it requires shipping the parts in bundles, thereby making it necessary to rely upon the dealer, or oftentimes the farmer, to sort out the pieces and attempt to fit them together the way the implement engineer designed them. Because manufacturing operations are not infallible, this procedure can result in improperly fitted parts and a resulting weak and inadequate structure.

So far, we have studied the influence of the front hitch upon the configuration of the tractor and some of the problems resulting therefrom. Now that we have the structure established it seems logical that we examine its market limitations as a result of raising the tractor and placing the wheels in close proximity. Because of the increased height, the sales department demands that the manufacturer produce a revised model having a lower silhouette for orchard work. In addition to this second model, another model with reduced height, and embodying a pivoted axle with wheels placed at each extremity, must be offered to capture a part of the industrial market. A no less important factor is the limitation of materials handling work by the tricycle tractor in normal farm operations because of reduced flotation and stability. With the advent of front-and-rear loader equipment, an ever increasing demand will be realized for a tractor capable of handling these tools as well as those required for general farm work.

Unlike the previous section of the paper in which it was indicated customer preference justified the front-hitch design, even though many compromises must be made, this section will attempt to point

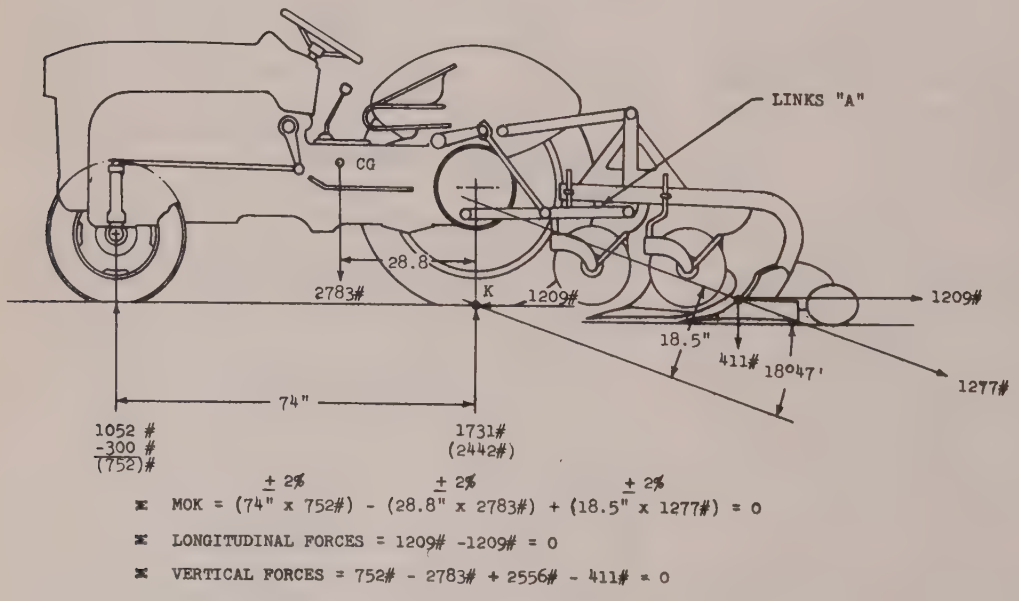

Fig. 1 – Resultant force diagram with 2-bottom. 14-in., general-purpose lift-type, moldboard plow at 2.3 mph – furrow depth 6.5 in., furrow width 29 in., wheat stubble

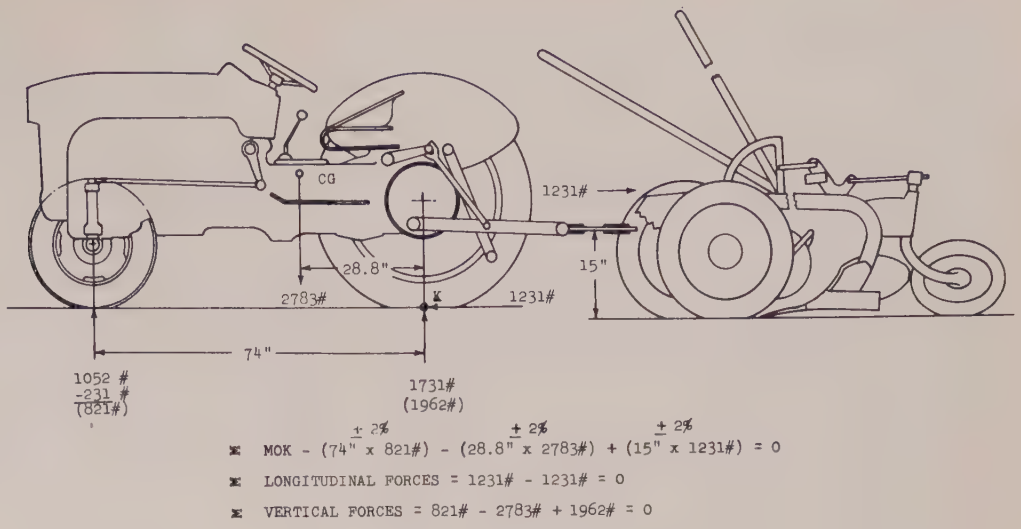

Fig. 2 – Tractor linkage loads and weight transfer on front axle in plowing with 2-bottom, 14-in. wheel plow at 2.3 mph – furrow depth 6.5 in., furrow width 29 in., wheat stubble

out the reasons why demand for trailed implements is rapidly diminishing except on the very large models of tractors. As a post-mortem to this evolution from the early steam tractor, it seems justifiable to examine the basis for this trend, and in turn to discuss the functional characteristics and limitations of both types of hitches, as well as their effect upon the design of the tractor and implements.

Because there are several types of rear-hitch designs for integral tools on the market, most of which have operating characteristics unknown to the writer, this section of the paper, of necessity, is limited to the 3-point draft control type of hitch as used on the Ford tractor, as compared to the conventional hitch for trailed equipment.

It is the writer's belief that maneuverability and flexibility of operation transcends all other factors in bringing about the increased demand for tractors equipped with the integral implement hitch. As highways have cut inroads through the rural communities, it has become more and more necessary for the farmer to cross these arteries of transportation to contact all sections of the farm with his equipment. Under these conditions the farmer saves many hours each year by carrying the implements on the tractor instead of by trailer or other means. Furthermore, the integral implement hitch broadens the scope of usefulness of the tractor, inasmuch as it permits using such special equipment as blades, scoops, portable saws, post-hole diggers, brushes, and others too numerous to mention. These many devices have extended the tractor's useful work period from a seasonal to a 12-month operation. Many of these tools have also provided a means for the purchaser to do custom work during slack periods, thus increasing his return on investment. The problems of working fence corners and marginal land with trailed equipment no longer exists with the advent of integral implements. While this is not a serious problem on large acreages, it becomes an all-important factor on small farms and garden plots.

One of the basic differences between the Ford tractor with its mounted tools and a tractor with trailed implements is the weight required to prevent excessive wheel slippage during normal tillage operations. The linkage and controls on the Ford tractor are designed to provide weight transfer from the implement to the tractor, thereby reducing the vehicle weight required for normal operation. This is accomplished by establishing an unbalanced line of draft on the implement in respect to the hitch points on the tractor, and in turn balancing the forces by a hydraulic system.

To understand this better refer to Fig. 1, in which is shown the tractor pulling a mounted 2-bottom plow. In this sketch it should be noted that the tractor, less plow, weighs 1052 lb on the front and 1731 lb on the rear, for a combined weight of 2783 lb. The plow weighs 447 lb. When plowing, the weight of the plow, plus the vertical reaction because of suck into the soil, are supported by links "A," which in turn are supported by a ram cylinder in the hydraulic system. As a result of these factors, additional weight is transferred to the tractor chassis from the implement. The lower figures, shown in brackets, indicate the tractor weight on the wheels while operating under average conditions. It should be noted that the weight on the drive wheels has now increased from 1731 lb to 2442 lb. This increase results because of the aforementioned conditions of supporting the implement and as a result of the normal weight transfer from the front wheels to the rear wheels when pulling from a point above the contact line of the rear wheel and the surface upon which it rolls.

Fig. 2 shows the same tractor pulling a conventional trailed plow. In this case the increased weight of 231 lb on the drive wheels results entirely from the transfer of weight from the front wheels, therefore the total tractor weight has not increased.

In comparing the two methods of hitching, it can readily be seen that, under normal draft conditions, it would be necessary to add the difference of weight to the tractor with the trailed implement in order to maintain equal wheel slippage of the two units. If we consider that the average tractive efficiency of the tractor drive wheel on stubble fields and plowed ground is approximately 60%, it becomes apparent that every pound of draft of the trailed tool requires 1.66 lb of weight on the drive wheels of the tractor to propel the implement. Therefore, tractors equipped for trailed equipment must be burdened by excessive weight that, unfortunately, cannot be removed for light draft operations.

A comparison of tractors in the full 2-bottom plow class indicates additional built-in weight in excess of 1000 lb for units trailing plows, as compared to the Ford unit. A complete paper could be written on the initial cost and upkeep of this additional 1000 lb, however, space will only permit reviewing some of the more important problems as related to weight and hitches.

The most serious consequence of excessive tractor weight is its adverse effect upon flotation and soil compaction. The tractor equipped with hydraulic draft control and mounted implements has the ability to tread lightly when necessary in places of marginal flotation because of the draft compensating mechanism in the hydraulic system and the fact that the implement depth is by draft reaction instead of by gage wheels. With the mounted implements the operator can, at will, raise the tool out of the ground, back up, and change his course of direction when necessary. With trailed equipment it is often necessary to disconnect the tractor from the implement, drag the implement out on firm soil or away from its obstruction, and rehitch.

A further objection to this excessive tractor weight, because of trailed implements, is the power requirements to propel the additional weight. The horsepower losses, of course, depend upon many design and operational factors; however, every increase in weight results in a proportionate amount of horsepower loss and should, therefore, be given serious consideration.

Now that we have reviewed some of the effects upon the tractor as a result of applying the two hitches, we should consider two important limitations of the trailed implements. Because the hitch requires a much heavier tractor to provide draft without excessive wheel slippage, the dynamic loading upon the equipment, when striking obstructions, is much more severe than the loads imposed upon mounted equipment. As a consequence, the implement must be designed heavier to withstand the higher impact loading. The implement designer is also perplexed by the problem of not knowing what size tractor may be used for the draft medium. Because of this the implements must be overdesigned. Trailed implements are then burdened by additional cost, both because of excessive tractor weight and because of gage wheels, axles, and adjustment mechanisms.

It has often been said that the problem of implement interchangeability with mounted equipment is a serious barrier to sales and one that does not exist with towed implements. However, if we consider the many advantages accruing to the well-balanced tractor and implement combination it becomes apparent that the gains overshadow this objection. In this era of specialization it requires that compromises be limited when attempting to accomplish the objective of faster and more efficient methods of production. Because the implement and tractor design are integrated with the mounted hitch, the package can be designed with maximum efficiency. It no longer becomes necessary to overdesign the units because of a lack of knowledge as to the loads that may be imposed upon the structure.

It should be recognized that, because of this fundamental concept of balance, the integral implements must be limited as to size and design, otherwise serious conditions of instability and malfunctioning will result. This is particularly a problem with tractors embodying draft control hydraulic mechanisms, such as the Ford, inasmuch as the tractor must be sufficiently stable to resist the dynamic forces imposed upon it by the implement. Because the tractor's hydraulic mechanism maintains a relatively uniform load upon the implement, it serves to prevent overloading of the tractor due to rapid changes of draft. It also permits operating the tractor over undulating surfaces without adversely affecting the implement position in relation to the surface. This is because the implement position is automatically determined by draft reaction on the top member of the linkage rather than by a fixed linkage on the tractor chassis.

Since draft reaction on the top link is unobtainable with certain pieces of equipment, and because, under certain conditions of operation, depth must be maintained irrespective of draft, we have found it necessary to provide a second system of control in order to maintain the implement in direct relation to the tractor chassis irrespective of draft. The two systems of control can be selected at will by the operator rotating a small selector lever located beside the seat. These two systems of control have been pointed out in order to indicate that one type of control was found to be inadequate, and the problem of mounting tools on the tractor embraces more than a set of links and a lift. It is only logical to believe that each design of linkage and controls furnished on the various makes of tractors contains operational characteristics peculiar to the design.

These characteristics must be thoroughly understood by the implement engineer before he can objectively approach the problem of supplying a design compatible with the tractor. The problem is compounded because of the different type of hitches now being used. While it would be desirable for

the farmer and implement manufacturer to have a common hitch on all makes of tractors, it is questionable whether this could be accomplished without similar methods of hydraulic control. However, it is within the realm of possibility that some standard may eventually evolve from the many designs now available, after the industry recognizes the best approach to the problem. In the interim period it would seem good business prudence to make available throughout the industry the basic information necessary for the implement and equipment engineer to design suitable tools to work on the various hitches.

A sample of such information is shown in Fig. 3, wherein basic information relating to the hitch and capacities of the hydraulic circuits is shown. It should be recognized that, even though the implement matches perfectly with the hitch and is within the hydraulic capacities of the circuit, there still can be no assurance of satisfactory operation without extensive field testing.

As a final consideration we should examine the provisions necessary in the hydraulic system and hitching members in order to fulfill the requirements of the various implements and equipment. As mentioned previously, our experience indicates that the hydraulic system should provide control both by draft reaction and by selective positioning of the tools in relation to the chassis. Furthermore, the system should provide a means of adding additional remote circuits for operating both single-acting and double-acting cylinders. The circuit

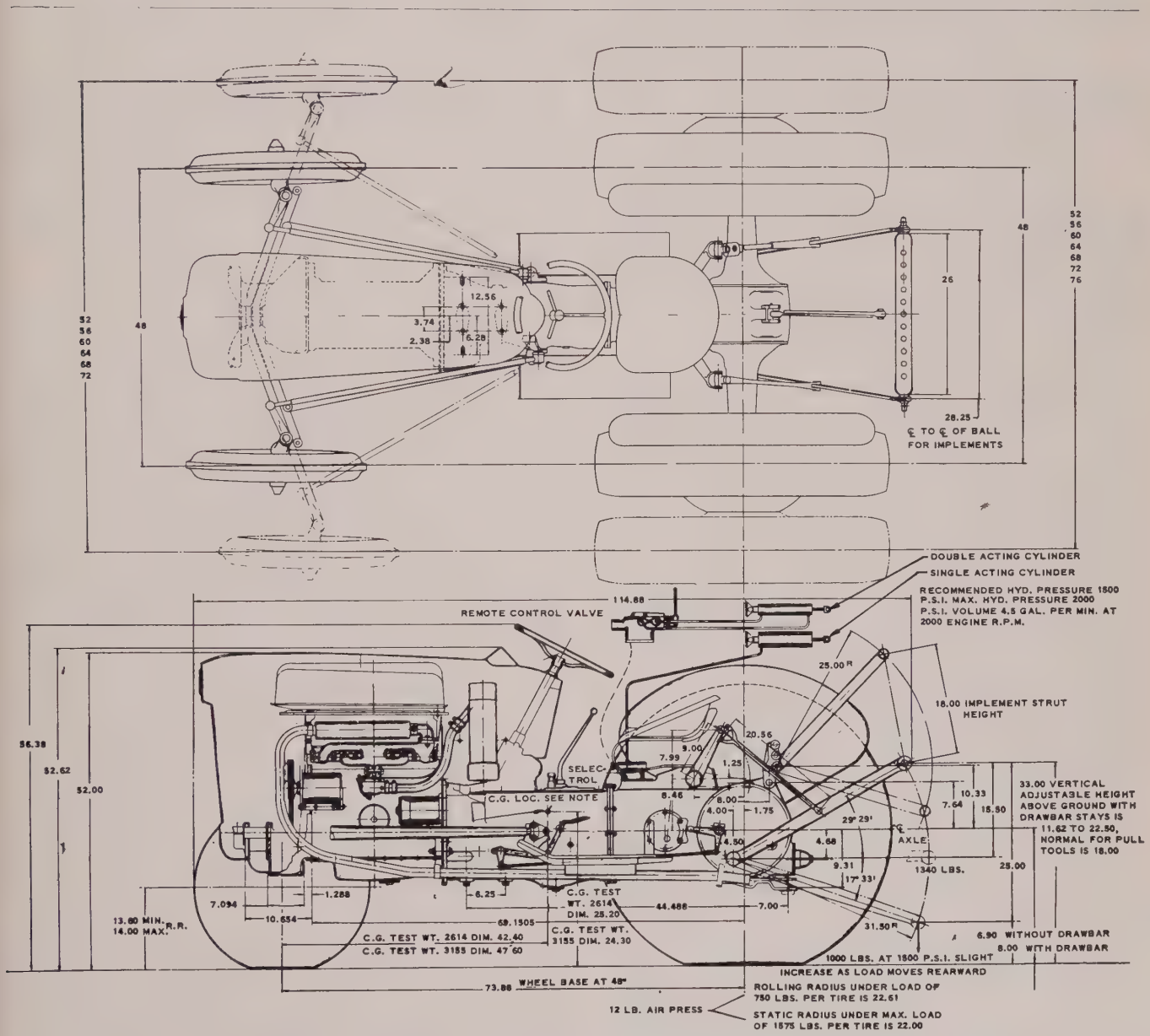

Fig. 3 – Basic information relating to hitch and capacities of hydraulic circuits

capacities should be adequate to fulfill the SAE requirements for remote cylinders. The hitch should provide ample support without requiring excessive framework in order to stabilize the implement.

The hitch should provide for longitudinal and lateral adjustments. It should also permit lateral movement for contour work as well as a means for immobilizing this motion when required. The hitch should provide implement trailing characteristics that follow the direction of steering as well as stabilizing the implement on hillsides. The linkage should approach a parallelogram in the vertical plane in order to minimize the effect of overhung implements on hydraulic lifting capacity. Motion of the implement should be restricted in the transport position, in order to reduce damage to the tractor and equipment when operating over rough surfaces at high speeds.

Although the hitch and hydraulic system are but two of the many components making up the tractor, it becomes apparent that they dictate the general design of the vehicle. Hence, careful analysis of the effects upon the tractor must be made before launching a design program. We have reviewed the three current positions of hitches and some of the problems confronting the designer as related to each. While this paper has not attempted to point the direction for future designs, it is hoped that the information presented will serve as the warp upon which can be interlaced the threads required to complete the picture of the most desirable type of hitch and hydraulic system.

DISCUSSION

Points Out Advantages of Front-Mounted Equipment

– Igor Kamlukin
Allis-Chalmers Mfg. Co.

WE agree wholeheartedly with most of this paper. We fully agree on the merits to be gained by the use of draft control for traction boosting, and with the benefits to be gained by use of integral-mounted equipment. However, we believe that more could be said for the front and rear integral-mounted equipment.

We do not wholly agree with Mr. Brock that the relative merits of front-rear integral-mounted cultivators versus rear integral mounting can be dismissed simply on the basis of personal preference. Front-mounted equipment is, first of all, less fatiguing on the operator because the operator does not have to keep his head turned to see that his equipment is working properly and not getting plugged with trash. On hillsides, front-mounted equipment does not drift downhill and allows closer cultivation. Also, when used on contoured land, front-mounted cultivators are easier to guide. It is because of the wide demand for front-rear integral cultivating equipment that the designer is forced to look for satisfactory solutions to overcome the difficulties enumerated by Mr. Brock. However, the solutions are not as difficult as intimated. A tractor designed for front-mounted equipment must have sufficient wheelbase to permit cultivating equipment to be placed behind the front and rear wheels, and this feature in turn moves the center of gravity forward of the rear wheels to give increased longitudinal stability, and the increased length permits placement of the fuel tank and battery behind the engine to keep the center of gravity low.

Final drives with smaller tires are another means of keeping the center of gravity down and at the same time providing ample crop clearance. The smaller tires also permit hitching rear-mounted equipment closer to the tractor thus permitting better implement control as well as permitting use of heavier implements without overbalancing the front of the tractor when in transport.

There is no question that the wide front axle is growing in popularity because of the increased stability, improved ride, and easier maneuverability possible with this design. However, to make this type of axle work properly with front-mounted equipment, it must be designed so that the area back of the axle in which the implements work is not blocked by the radius rod of the steering mechanism. The increase in popularity of the adjustable front axle does not, however, mean that the tricycle tractor is on the way out. Equipment like the mounted cornpicker needs the clearance that is blocked by the wide axle. Fortunately, it is possible to make a tractor in which both axles are easily interchangeable and thus realize the benefits of both.

Admittedly, the problem is more complex when integral tools have to be made to attach both at the front and rear of a tractor. However, every year improvements and simplifications are coming into existence to make it easier to do so. Front-mounted equipment is now designed so that a tractor can drive into it and couple up with a minimum of effort. Snap couplers are developed so that rear mounted equipment can also be coupled with virtually no effort. The need for easy adjustment of rear-wheel treads becomes more important with mounted equipment because, in effect, the rear wheels of the tractor become the wheels of the implement. As an example, a mounted plow requires a narrow tread so that it will make the proper width of cut. A mounted disc, which because of its weight must be coupled close to the tractor, requires a narrow tread to permit turning the tractor without cutting the rear-wheel tires on the discs. With a field cultivator a wide tread is necessary to stabilize the implement, and with a cultivator the tread adjustment must change with the crop row widths. Since many of these operations may have to be performed in the same day, the need for an easy means of changing the rear-wheel treads of a tractor becomes apparent. Demonstrations showing how fast one can attach different implements are meaningless unless the preparation of the whole tractor to work with those implements is also a simple and quick operation.

Regarding hydraulic controls, Mr. Brock points out there is a great deal to be gained by using the weight of the implement and transferring it to the tractor to have additional weight on the rear wheels for increased traction. During the development of the Allis-Chalmers hydraulic system to accomplish this weight transfer, we found that if the implement was pulled from a single point, we were able to maintain a more uniform implement working depth. Also, by using a single point of pull an appreciable distance above the ground, we were able to have some weight transfer when entering a field before the implement came to its full depth. This feature is particularly valuable when working hard soils. By using a single point of pull close to the rear axle, the plow or bedder can be beamed so that it floats at the desired depth, the hydraulic system is then adjusted to develop just a very light pressure in the lift-

ing cylinders. With this adjustment, when the implement enters soil that creates increased draft, the pressure in the lift cylinders increases and the portion of implement weight resting on the ground can be picked up by the tractor and used for weight transfer to increase traction. If the load increase is not great enough to demand a weight transfer greater than that resting on the ground, the implement continues at the depth at which it would normally float. Since in practice the draft load is constantly changing while traveling through a field, this lifting force is applied in pulsations and the actual amount of weight transfer possible is increased by the magnitude of inertia forces. To facilitate adjustment of the hydraulic system by the method outlined above, we have found it necessary to develop a high-pressure gage to show the pressure in the lifting cylinders and to show an operator graphically when the system is transferring weight for traction boosting.

As the author points out, the hydraulic system should be capable of adjustment for selective positioning of tools in relation to the chassis as well as be draft responsive. However, we have not found it practical to control plows, bedders, field cultivators, or subsoilers, or other similar tools by this positioning control, as up or down variations of the tractor change the position of the implement in the ground too much. We limit the use of positional control to such operations where the important consideration is maintaining a position of one member with respect to another, such as is necessary for use with loaders, cranes, mowers, cutter bars, and similar equipment.

More Suitable Front-Mounted Cultivator Mounting Is Needed

– George M. Eveleth
J. I. Case Co.

I AM in full agreement with the author's statement in regard to the trend toward the design of the tractor and implement as a unit for all row-crop-size tractors. I believe that most tractor companies now pretty much consider their general-purpose type of farm tractors as merely power units that can be used with various mounted tools to provide a self-propelled plow, self-propelled disc harrow, planter, cultivator, corn picker, mower, or any one of a great many other implements. The several new rear-implement hitches that have been provided during the past few years have greatly increased the usefulness of tractors by making it possible to change easily from one implement to another in a short period of time. This trend has not been limited to the smaller single- and 2-plow-size tractors as it is also being followed in larger row-crop units. The weight and size of the rear-mounted equipment will, no doubt, be the limiting factor in determining the largest size of tractors on which rear-mounted equipment with quick-change hitches can be successfully applied.

I must also agree with the author's statement in regard to the present need of close cooperation between tractor designers and implement designers. This need has certainly changed from the days of all pull behind implements in which the only connection between the implement and the tractor was the drawbar pin. This need of close cooperation between implement and tractor engineers is reflected by the presence of both implement engineers and tractor engineers at many of our technical society meetings regardless of whether these meetings stress tractor design or implement design.

I would like to say a few words in regard to Mr. Brock's comments on front-mounted cultivators. While I realize and will admit that some users can and do get by with rear-mounted cultivators only, I do know of many other users who insist on front-mounted cultivators for many cultivating operations. This is especially true on contour rows to facilitate following the crop rows. It is also desirable in many irrigated areas in which a combination front-and-rear-mounted cultivator can do the same work in one trip over the crop as is accomplished by two trips for a rear-mounted unit only. It is, therefore, economically necessary for such farmers to use equipment that enables them to eliminate the time and expense of passing over their crops twice when the same work can be accomplished with one cultivation.

I do think that we farm machinery manufacturers must give more thought and effort to provide a more suitable front-mounted cultivator mounting—one in which the cultivator can be attached or removed from the tractor as quickly as our present rear-mounted equipment. When a front-mounted cultivator that will provide satisfactory field performance is developed that can be mounted on the tractor or removed from it in a matter of several minutes without special handling facilities, I am sure we will find a greater number of farmer users who will prefer this type of equipment to a rear-mounted cultivator only.

I am also inclined to feel that Mr. Brock has exaggerated in regard to the need of raising the center of gravity of tractors on which front-mounted cultivators are mounted. It is my opinion that there are several tractors in the 2-plow-size class that can accommodate front-mounted cultivators and these tractors have a center of gravity within 2 to 3 in. of that illustrated in Fig. 3 of his paper. Therefore, I think that he has over emphasized the loss of longitudinal and lateral stability of tractors designed to accommodate front-mounted cultivators.

While his paper does not specifically indicate, I was given the impression that the author assumes that front-mounted cultivators are associated only with the tricycle- or front-pedestal-type tractors. I know of some popular tractors furnished with adjustable front axles and on which front-mounted cultivators can be used with the same beneficial results as pointed out by the author for a 4-wheel tractor.

He indicates that the rear hitch should provide lateral movement for contour work. This type of connection might be acceptable for many operations; however, there are other operations in which the lateral movement is used to considerable disadvantage. For instance, in contour moldboard plowing when making furrows on an outside curve the plow will overcut by as much as the limiting device will permit. This means that all the soil is not being plowed. On inside curves the plow will undercut by the amount that the limiting device will permit, which means that the user is not plowing the same width as he does when plowing straight furrows. Neither of these conditions results in a good job of plowing, especially in contour strip farming where it is desired to maintain uniform width of strips. It is my opinion that a hitch connection and plow assembly should be provided in which the width of plow cut is quite uniform regardless of whether the tractor and plow are used for straight furrows or contour furrows.

The author points out that draft control for rear-mounted implements is not adequate for all conditions and that it is necessary to provide a second system of control in order to maintain proper depth of operation for certain implements. Personally, I am not a supporter of draft control as it seems to me that it is merely a means of "short-changing" the working depth of the implement to permit the tractor to continue operating without shifting to a lower speed. In other words, it is a means of relieving the load on the tractor by sacrificing the quality of the work being done. I do not think that the average farmer is willing to sacrifice on the quality of the work being done in order to permit the tractor to continue operating at the same load when he encounters some tough spots in his fields. It is usually those tough spots where the full depth of penetration is most desireable from the standpoint of quality operation.

We should all be in agreement that the hydraulic system should provide a means of adding remote circuits for actuating remote hydraulic cylinders in compliance with the

standards on which we have already agreed. Therefore, I think that any present-day agricultural tractor is not complete without a means of accommodating the recommended hydraulic cylinder for remote application.

Stability Problems Plague Designers Of Large, Rear-Mounted Tractors

– Thomas Evans
Minneapolis-Moline Co.

THERE can be no doubt that Mr. Brock's concept of the integral unit of tractor and implement each designed to fit the other for maximum economy, both from an operational and a material point of view, is something for which the farmer-customer is going to maintain a continuing demand. As the author has clearly set forth, the problems of road and highway transport are something that have to be faced by the implement designer, who might well use the wheels and tires already supplied on his mobile power unit, the tractor, without resorting to auxiliary wheels, tires, and necessary structure to support these transporting means. The question of keeping the implement in one piece when dismounted from its power source is certainly one that is important from a time and storage point of view.

It is because of these and other reasons that I would like to explore the problems of tractor hitches and their effects on tractor configuration when applied to tractors that are larger than the 2-plow tractor that Mr. Brock has used in his comparison of hitches and the problems of tractor configuration when these larger tractors are used as an integral unit with mounted harvesting machinery. It is not the intent of this discussion to compare the actual characteristics or the merits of existing hitches now in use for mounting integral implements, but rather to attempt to show that in a more general solution there are problems involved that make it doubtful if rear-mounted hitches on the conventional tractor configuration fill the bill for all-around use. Moreover, no attempt has been made to predict what the general solution might be.

There has been, up until now, a noticeable upward trend in the amount of power contained in a single farm tractor and the participation of the larger power units in the total market has also shown a marked increase over the years. Unquestionably, at least one of the factors contributing to this trend is the economic consideration of putting as much power as possible at the disposal of a single operator. Whether this trend will continue is conjecture, but there is no reason to believe that it will not. Since the virtues of integral equipment seem to apply to these larger power sources as well as to the small, it would be well at least to list some of the problems arising from this consideration.

Let us first examine the problem of the larger tractor equipped with rear-mounted tillage tools. In general, tillage implements as they are currently conceived, are groups of comparatively small ground engaging tools. As the size of the implement is increased, the size of the unit tool in general does not increase proportionately, but more of the units are used in the group. These units are disposed in two main ways, namely in echelon formation such as is found in moldboard and disc plows and in laterally deployed formations such as is found in middlebreakers and planters. Combinations of the two methods are, of course, apparent in cultivators and disc harrows.

When considering the rear-hitching of completely mounted tillage tools of the echelon type in sizes larger than those used with 2-plow tractors, we find that a major problem of longitudinal stability of the tractor with the implement in the transport position becomes immediately apparent. The draft requirement and therefore the weight of the tractor need be only proportionate to the number of units, but the weight of the implement together with its center of gravity location behind the hitch points, both vary with the size of the implement so that the overturning moment of the tractor increases geometrically. It, therefore, appears that changes in configuration of the tractor or new conceptions of the disposal of such tillage tools become necessary if we are to continue to hitch these tools at the rear of the tractor. Another factor in the echelon formation, when considered from the point of view of a draft sensitive upper link of the 3-point linkage hitch such as the author has discussed, is the fact that the force in the upper link tends to diminish with each additional unit until it is questionable that it would adequately measure the draft requirement at all.

When considering the rear hitching of completely mounted tillage implements that are in transversely deployed arrangements, we find that the problems of the lateral working stability of the implement in the ground transcend those of draft regulation and weight transfer so that any hydraulic control, whether it be manual or automatic, should be applied to this function. The larger implements of this type would normally need a more widespread base to support and control them to overcome this stability problem than would the 2-row implements of the type generally employed on the 2-plow tractor.

Another major factor to be considered when discussing the hitching of the larger tillage equipment is the question of the act of physically making the connection between tractor and implement. When the tool is too large to move manually to compensate for minor deviations in backing up to it, some mechanical means of making this compensation has to be incorporated.

In order to make the tractor a completely versatile power unit we should also look at its configuration from the point of view of harvesting equipment rather than only from that of the tillage tools. Harvesting machinery of the tractor mounted type as currently conceived does not lend itself well to rear hitches on a tractor of conventional configuration. The successful attempts that have been made to arrive at a tractor configuration that is versatile for integrally mounted harvesting equipment is markedly different from that employed for tractors intended for tillage tools. It would appear that, until the tractor can be used completely through the season from ground preparation and including harvesting, we have not yet secured the ultimate evolution in tractor or hitch arrangement. This problem, compounded with the problems involved with the larger tractor, appear to leave the ideal arrangement, as yet, unsolved.

In summary, therefore, it is this writer's view that there can be no "standard" hitch or hydraulic system until the problems of hitching and controlling the larger mounted implement are solved. These implements should include harvesting equipment as well as tillage tools so that the tractor would be completely versatile throughout the farming operation.

Claims Tricycle Tractor Is the Most Versatile

– Clarence A. Hubert
International Harvester Co.

THE author is certainly correct in his statement that present-day farm management is cost conscious. The farm operator is no longer plagued with fears of machine shortages. He is again in the position of deciding values relative to available types of tractor hitches and implement systems and may ultimately demand a standard arrangement of implements and tractors, despite conflicting theories and arguments of professional engineers.

We have long recognized the demand for tricycle and wheel tractors alike to satisfy the requirements of varied farm and industrial operations. The quantities of Farmall tractors built and sold, including the new Super H and Super M models, run well into the millions. It is, there-

fore, apparent that the customer would not be happy with the idea of having to abandon his high-clearance row-crop machine that so universally plows, plants, cultivates and harvests to his satisfaction. It is also interesting to note that a variety of front-axle attachments are also sold for these very same models and that for certain applications they are interchanged during the season.

Statistics reveals that the Farmall tractor, and the many copies of this prototype produced throughout the industry, is a most desirable tool. In conjunction with hydraulic controls, which were made available for certain farm implements long before 1939, the tractor successfully handles front-and rear-mounted implement attachments and affords the operator selective- as well as delayed-lift action. The inherent undercarriage clearance affords ideal row-crop clearance for 2-row cultivation.

Surveys have repeatedly shown that farmers prefer front-mounted cultivating equipment over the rear type. However, as Mr. Brock points out, the Ford-type tractor is not adapted for front-mounted tools and evidently certain front-mounted equipment applied to this tractor has not been very successful due to the double-draglink type of steering and lack of fore-and-aft clearance. As stated by the author, this is due to the fact that the tractor was not originally designed for this desirable feature.

Owners of the tricycle tractor will tell you that they sit comfortably well above the field dust-cone during their working day. While they cannot expound regarding the summation of forces and moments about the drive wheels, they will, however, tell you without hesitation about the pulling effect of the tractor with the use of wheel weights and the lower plowing cost per acre. Incidentally, several well-known attachment makers supply front- and rear-wheel weights for the Ford and Ferguson tractors. It, therefore, appears that this type of tractor uses its share of wheel weights.

The steering characteristics, flotation of tires, and front bolster structures of most tricycle tractors have been well accepted by the farmer. The steering-gear design on this type of tractor involves no more problems than on any other type. Steering is accomplished by the proper choice of mechanisms.

When heavy equipment such as front-end loaders are carried on a front hitch, the dual tires follow the ground level of the working bucket. This condition is ideal and not easily duplicated on a wide front-axle wheel-type tractor. In the 4-wheel arrangement, the front steering wheels normally lie outside the bucket dimensions and hence do not follow the ground level cleared by the bucket when it is pushed into the load.

The entire tractor industry is well aware of the demand for a clean plowing job in the field to produce a furrow of constant depth. This demand undoubtedly has had some influence in augmenting the former draft-control hydraulic system by the development of position control that more nearly approaches the hitch point control type. We must not forget, however, that first-grade plowing has been attained with the use of trailing implements and is still the criterion by which results of mounted plows are measured.

We have long recognized the desirable features and undesirable penalties imposed by the many single-point drawbars and three-point hitch systems. We have known the advantages and disadvantages of certain trailing implements and their counterparts, the mounted implements. While it is recognized that all engineering, including the arts of agricultural machinery, must represent certain compromises, the important point to determine is how far good field-working tools (providing the most proficient methods of tilling the soil) are to be compromised in order to maintain a specific mounted-type implement and draft-sensitive hydraulic system, originally conceived years ago for one principal purpose, that of plowing.

Many special features in the Ford-type tractor are recognized for medium farming and industrial applications, and parts of the implement lift system are indeed novel and a credit to the ingenuity of the designers. I believe, however, there will be an increasing need for the tricycle tractor with its versatile roving-type double- or single-acting hydraulic system, its wide use of front-mounted implements with improved attaching means, and a system such as the new "Fast-Hitch" now available on the Super C tractor.

I believe this statement is readily borne out by the fact that the tricycle design, with its quickly and easily interchangeable dual wheels, single-wheel, and adjustable wide front axle, can handle with ease the various machines that are now pulled or self-propelled.

The complex hydraulic systems and other new developments that we will see in the next few years must not be considered alone in the light of engineering achievements. They must be evaluated coldly on a dollars-and-cents basis, and very little has been gained if the farmer cannot mount such machines as corn pickers, cotton pickers, and the many other large and expensive machines and thereby increase the revenue he receives from such a tractor.

The tricycle tractor is most versatile, and I believe it will continue to be the criterion by which other types are measured.

ORAL DISCUSSION

Reported by A. C. Cragg
Donaldson Co., Inc.
and
Carlton L. Zink
Deere & Co.

W. E. Swenson, Minneapolis-Moline Co.: In ordinary plowing, how often is it necessary to adjust tilt of plow through the hydraulic system?

E. W. Tanquary, International Harvester Co.: Not often, in ordinary plowing. Generally, the wider the tool, the more frequent the adjustment.

B. G. Van Zee, Minneapolis-Moline Co.: In the interests of safety, how high can the drawbar be raised?

Mr. Tanquary: Approximately 22 in., which is about the same as for a standard drawbar. Most hydraulic drawbars lock in the "up" position for the sake of safety.

C. L. Zink, Deere & Co.: What are the requirements for and effectiveness of the double-action feature of the Super C hydraulic system?

Mr. Tanquary: The double-action feature was first used for the remote cylinder control of trailing implements, such as disc harrows and plows. On the Super C the remote cylinder is used to raise and lower rear-mounted implements. The shift of weight of the rear axle is a distinct advantage on moldboard plows. The double-action feature may be used to provide down pressure on rear cultivator rigs, disc plows, and other implements or, at the option of the operator, the tools may be allowed to float.

C. N. Hinkle, Standard Oil Co. (Ind.): Hydraulic engineers should give every consideration to the use of a hydraulic fluid that is readily available in the development of new hydraulic systems. If unobtainable hydraulic fluids are specified, the oil companies have found from experience that the farmer will use readily obtainable motor oils of light viscosity. Most motor oils have added to them an anti-foaming ingredient, and it should not be necessary to make such extreme demands upon the hydraulic system that a special fluid is required.

C. A. Hubert, International Harvester Co.: All discussions indicate that versatility, with respect both to various tractor models and to implement applications, is a primary requirement. I endorse this belief. Moreover, because of the increasing cost of the hydraulic system, it must have the widest possible application. Cylinder size, pump capacity, and other similar components should be subject to modification so as to meet new requirements or better handle the old ones.

Forty Years of Progress in Automotive

THE "forty years" in the title of this paper is intended to represent a long time rather than a specific number of years. In 1921,[1] B. B. Bachman, in discussing the standardization work of the SAE, said:

"The old Association of Licensed Automobile Manufacturers was formed early in the history of the industry, but produced no real cooperation as realized today. Associated with this organization was the so-called Mechanical Branch . . ., and while some progress was made in exchanging information . . . the suspicion of the other fellow and his motives prevented a realization of all the benefits of cooperative endeavor.

"Although this means of interchange of ideas existed, there were men whose vision reached out to a broader type of organization and a few of them got together and effected an informal organization in 1904 followed by the adoption of a constitution and the election of officers. They were: A. L. Riker, president; Henry Ford, first vice-president (and others . . .).

"From this beginning of the Society of Automobile Engineers, there was a gradual growth until in 1910, after the election of Howard Coffin as president, Coker F. Clarkson was secured as secretary and general manager and the beginning of a permanent office force effected."

Mr. Riker continued as president through 1907 and Henry Ford continued as vice-president through 1908. In 1917 the name of the society was changed from the "Society of Automobile Engineers" to the present name, "Society of Automotive Engineers, Inc."

From this record of the origin of the SAE it is seen that the SAE is almost 50 years old, but I have not been able to learn much about the progress in automotive lubrication during the first ten years, so perhaps the "forty years" in the title of this paper is satisfactory.

SAE records indicate that the first SAE Fuel and Lubrication Division of the SAE Standards Committee was organized Sept. 10, 1917, with H. L. Horning as chairman. In 1920 it was reorganized as the Lubricants Division of the SAE Standards Committee. Then in 1946 it became the Lubricants Committee of the SAE Technical Board, in 1947 it became the Fuels and Lubricants Committee of the Technical Board, and it is now called the Fuels and Lubricants Technical Committee. The chairmen and vice-chairmen during these years are named in Table 1.

From 1921 to 1946, inclusive, for 26 years, I was either chairman or vice-chairman. Since I had been around for this length of time, and since Father Time has finally caught up with me and I have reached the retirement age, your program committee figured that a brief review of progress in automotive lubrication might be in order. In regard to the "Look into the Future" part of my

[1] See *SAE Journal*, Vol. 9, July, 1921, pp. 33-34, 38: "Standardization as Brought about by the Society of Automotive Engineers," by B. B. Bachman.

THIS review of early developments in automotive equipment and its lubrication requirements was presented by Mr. Mougey upon the occasion of his retirement from General Motors.

He covers such subjects as crankcase oil specifications, crankcase dilution and corrosion, oil consumption, break-in oils, e-p crankcase oils, hydraulic valve lifters, relation of oils to octane requirement increase in engines, and hypoid-gear rear-axle lubricants – discussing what progress has been made and what problems are still to be solved.

He also pays tribute to a few of the many individuals who contributed to the progress that has been made in automotive lubrication. He points out that there are many other subjects on which progress has been made, but that he couldn't cover them all in one short paper.

Lubrication—A Look into the Future

H. C. Mougey, formerly with Research Laboratories Division, GMC

Paper was presented at the SAE National Fuels & Lubricants Meeting, Chicago, Nov. 5, 1953.

assignment, that is a more difficult matter. C. F. Kettering, who for many years was general manager of the General Motors Research Laboratories Division, has spent a lifetime in research and development work, and perhaps we may profit by examining his methods. In the 1924 annual number of *Motor,* in which all the new cars exhibited at the national automobile shows are described, we find a prediction by Mr. Kettering under the title, "The Car You'll Drive 10 Years from Now."[2] In this 1924 prediction Mr. Kettering said:

"It is a pretty hard job to make prophecies in the automobile business, not because certain tendencies cannot be seen that are bound to result in important changes, but because a man hesitates to tell all the possibilities; most people will not only disbelieve this, but will think he is a candidate for the asylum. . . . The important thing is that we have learned to wish for something unreasonable – and then go out to make the unreasonable thing something that everybody can have as a matter of course . . ."

Among the illustrations given by Mr. Kettering we find the following: ". . . We have learned a little something, now, about how combustion takes place in a cylinder, and from research along that line we have produced certain materials which can be added to gasoline that will entirely change the nature of the combustion that takes place in the cylinder, and by the use of this new material we can absolutely eliminate the knock from motors. . . . In connection with research on paint . . . We have the paint; we can positively guarantee that you can leave your car standing out all year . . . and at the end of the year it will look the way it did when you bought the car."

Of course, most of the people who read Mr. Kettering's prophecy in 1924 did not believe what he said, but when we consider the automobile industry as it was in 1924 and then look at the developments that have taken place since then, we realize how good a prophet Mr. Kettering really was, and it appears that Mr. Kettering's method of attack has considerable merit.

Recent advertisements of the American Petroleum Institute, such as the one which is now running in current issues of the magazines, indicate that the oil industry is following the same method as that outlined by Mr. Kettering. The last paragraph from the advertisement now being featured by the American Petroleum Institute is as follows:

"Because they were free to try the impossible again and again, oilmen, like the men of aviation, have served you and the nation well in the 50 years since Kitty Hawk. They will continue to search out new horizons for you so long as this great American freedom exists." – *Collier's,* Sept. 4, 1953.

In this paper I will try to give a very brief review of what has been accomplished along various lines, and then suggest some of the problems. Whether or not we solve the problems will depend very largely upon how hard we work on them, and upon how much we really want the results, as measured by how much we are willing to pay in time and money to obtain the results.

Crankcase Oil Specifications

The first SAE specification for crankcase oil that I can find is in the ALAM Bulletin No. 24-A, Janu-

Table 1 – Chairmen and Vice-Chairmen

Year	Chairman	Vice-Chairman
1917 – 1919	H. L. Horning	. . . . .
1920	A. P. Eves	. . . . .
1921 – 1924	H. C. Mougey	W. E. Jominy
1925	H. C. Mougey	W. S. Cochrane
1926	H. C. Mougey	W. H. Graves
1927	W. S. James	H. C. Mougey
1928 – 1946	E. W. Upham	H. C. Mougey
1947 – 1948	E. W. Upham	M. D. Gjerde
1949 – 1950	M. D. Gjerde	J. M. Campbell
1951 – 1953	M. D. Gjerde	V. G. Raviolo

[2] See *Motor,* Vol. 41, January, 1924, pp. 84-85, 300: "The Car You'll Drive 10 Years from Now," by C. F. Kettering.

The Author

H. C. MOUGEY (M '20) is now retired from the General Motors Corp. where he had held the position of Technical Director of the Research Laboratories Division. After graduating from Ohio State University with a B.S. degree in chemical engineering in 1911, he joined the Lowe Brothers Co. In 1917, Mr. Mougey was associated with the Dayton Engineering Laboratories, which in 1920 became part of GMC.

ary, 1907, p. 213, Second Annual Specifications for Materials, by Henry Souther. Fig. 1 is a photographic reproduction of this same specification as it appears in SAE Transactions, Vol. 5, 1910, p. 174. The SAE records show that this specification was officially adopted in June, 1911. I believe this specification was the only SAE crankcase oil specification until March, 1923, when a series of specifications was adopted. However, since these new specifications covered a variety of properties in addition to viscosity, they did not come into general use.

In July, 1926, the first SAE crankcase oil viscosity-number classification was adopted. It was based on viscosity at 130 F for the lighter oils and at 210 F for the heavier oils. With a few slight modifications this SAE crankcase oil classification remained in effect until it was revised in October, 1950; at which time the "W" oils, 5W, 10W, and 20W were adopted.

The SAE number for lighter oils is now based on viscosity at 0 F, but for the heavier oils it is still based on viscosity at 210 F.

It is of interest to note that, although the 10W and 20W oils have been in commercial use for about 20 years, they were not officially recognized by the SAE until October, 1950.

In this connection, one of the most important developments in the past 40 years is the change in mental attitude that has occurred, largely within the last 15 years. A number of years ago many people believed that the lubricants desired could be obtained by specifying a list of properties, such as gravity, viscosity, flash, fire, and color. At the present time it is recognized that, although these various properties are of importance, their principal value is in checking the identity of various oils and determining whether or not an oil is similar to a previous oil whose value has been determined by actual tests. The real quality of an oil depends upon the performance of the oil in actual use, and this can be established much better by actual performance tests than by checking the oil against various items, many of which are not directly related to the manner in which the oil performs in actual service. The use of engine tests in the Army and Navy crankcase oil specifications and the use of rear-axle tests in the Army rear-axle lubricant specification are examples of this change in mental attitude.

This change to the appreciation of the importance of actual performance tests is really nothing new. We have always recognized that we can tell more about the desirability of an apple as an eating apple by actually eating the apple than we can by making such tests as weighing the apple, determining its diameter, circumference, and so forth. There are many common sayings illustrating this fact. For example, we say, "The proof of the pudding is the eating." Then there is the old rime about mushrooms:

"To tell a mushroom
Simply eat
The specimen
You chance to meet
And note next day
With studious care
If you've stayed here
Or gone elsewhere."

Crankcase Dilution and Crankcase Corrosion

Early issues of the *SAE Journal* contain many papers in which these problems are discussed. The heavy ends of the gasoline did not burn very well and they slid down into the crankcase, diluting the oil. The typical crankcase oil[3] in the average crankcase contained about 10 to 15% dilution in summer and 20 to 30% in winter. It also usually contained considerable water and other products of combustion of the gasoline, especially in winter or in cars used in light service, summer or winter. The heavy ends of the fuel reduced the viscosity of the crankcase oils to an alarming extent. To compensate for this dilution the public began to use oils of higher and higher viscosity, so that there would still be some viscosity left after the

[3] See SAE Transactions, Vol. 18, Part II, 1923, pp. 82-98: "Crankcase Oil Dilution," by E. F. Hallock.

AUTOMOBILE ENGINE LUBRICATING OIL.

SPECIFICATION No. 26.

Oil for this purpos must be a pure mineral oil, no addition or adulterant of any kind being permitted.

The following characteristics are desired:

Specific Gravity	28° to 32° Baumé
Flash Point, not less than	400° F.
Fire Test, not less than	450° F.
Viscosity at 100° F., Saybolt Viscosimeter, not over	300 seconds
Viscosity at 210° F., Saybolt Viscosimeter	40 to 50 seconds
Viscosity at 210° F., Tagliabue Viscosimeter	60 to 65 seconds
Carbon Residue, not over	0.50%

Fig. 1 – Copy of first SAE specification for crankcase oil

oil was diluted. However, the peak was reached[4] about 1932 when the average crankcase oil was about SAE 40. This fuel dilution problem was solved jointly by the oil industry and the automobile industry; the former by supplying gasolines that did not contain so much of the heavy ends, and the latter by using thermostats and crankcase ventilation.

However, the problem of crankcase corrosion, one of the earliest problems encountered in automotive lubrication, is not entirely solved yet, although the facts in regard to crankcase corrosion are well known and have been known for many years. Prior to 1924 it had been common experience to find water in used crankcase oils, but little was published on this subject prior to that date. At the Summer Meeting of the SAE in 1924, a number of papers were given on corrosion in automobile engines and on water in crankcase oils. I gave one myself.[5] However, at this same meeting A. Ludlow Clayden gave such a good paper[6] on "Water in Crankcase Oils" that his paper is the only one ever remembered; and even to this day, it is probably the best reference on this subject. Since this time literally hundreds of papers have been given on this subject. They may be summarized as follows:

1. Corrosion of piston rings, cylinder walls, and the like, due to water and acids in the water, is a serious cause of engine wear.

2. This corrosion and resulting wear can be decreased by the proper selection of fuel and by keeping the engine operating temperatures high enough to prevent condensation of the water and acids.

3. Where these favorable conditions cannot be obtained, crankcase oils that have the ability to "rustproof" the engine parts will greatly decrease the corrosion and resulting wear.

Problems Relating to Crankcase Corrosion

The first problem I would like to propose is that of developing a suitable test to evaluate crankcase oils in respect to their ability to decrease corrosion of the iron and steel parts and the resulting wear when the oils are used in service. It is generally recognized that heavy-duty oils, in general, are the best commercial solution to this corrosion problem at present, but should these oils, as supplied by the oil industry at the filling stations, be of the MIL-0-2104 level, or of the Supplement 1 level, or of the Supplement 2 level? Although the high-additive oils, in general, are very effective in decreasing corrosion and the resulting wear, are there some other materials, other than those now commonly used to give detergency properties, that can give equal or better corrosion-resisting properties to crankcase oils? Many men in the oil industry have objected to the use of the L-1 engine test in the evaluation of crankcase oils for gasoline engines. This is a diesel-engine test made using diesel fuel in a diesel engine, but, up to date, no one has developed a superior test for evaluating in gasoline-engine oils the properties of detergency. In addition, insofar as the detergent materials reduce ferrous corrosion due to water and acids, the L-1 diesel-engine test is at least a fair guide in selecting oils resistant to ferrous corrosion. A test that we use for research purposes to evaluate oils in respect to their ferrous-corrosion-resistant properties is shown in Table 2.

Table 2 – Special Test to Compare Oils in Respect to Their Ability to Decrease Rusting and Corrosion

Engine[a]	Chevrolet Powerglide
Speed, rpm	1500
Load, bhp	25
Water, In, F	85
Out, F	95
Oil, F	120
Fuel	Commercial premium gasoline containing 2.5 to 3 cc of tel
Air/Fuel Ratio	14.5/1 plus or minus 0.5
Time[a]	21 hr of engine operation on a cycling schedule of 3 hr running and 3 hr shutdown during 16 hr out of a 24-hr period. Engine is shutdown during the third 8-hr shift of each 24 hr

[a] The louvers in the rocker-arm cover are closed to decrease ventilation and thus to increase the tendency for rusting in this test procedure. New rocker arm shafts, intake valves, valve lifters, and piston pins are used in each test. The oils are compared by photographing the rocker-arm shafts, intake-valve stems, valve lifters and piston pins from each test. The intake-valve stems are soaked overnight in varnish remover to remove any varnish on the stems, so that the photograph will illustrate the amount of rusting and corrosion. The hydraulic valve lifters are disassembled after the test so that the photographs show the appearance of the hydraulic valve lifter pistons as well as the outside of the complete unit. If the original unit is "phosphate coated" the photograph of the "phosphate coated" part may be eliminated, as it would not be indicative of the amount of rusting and corrosion.

Since the nature of the gasoline used in making the test affects the results, the same gasoline should be used in tests comparing two different oils. In addition, if the test is not sufficiently severe, with the particular gasoline used, it may be necessary to increase the time beyond 21 hr of engine operation in order to obtain sufficient rusting or corrosion for purposes of comparison.

Oil Consumption

In the very early days of the industry the miles per gallon of lubricating oil were very low, even though average car speeds were rather low and, as is well known, in a car driven at low or moderate speeds the miles per gallon of oil will usually be much greater than in a car driven at high speeds. In the early days oil consumption was so bad that it was usually reported in miles per gallon instead of in miles per quart as is the practice today. C. W. Stratford in a paper[7] in the 1915 SAE Transactions says:

"The maximum mileage per gallon of oil in the best automobile motor is rarely above 1000 (250 miles per quart), as compared to that of the average motor, which probably never exceeds 200 (50 miles per quart)."

In the 1922 SAE Transactions F. E. Watts, in discussing a paper by A. A. Bull,[8] says:

"I believe that the ideal oil consumption at pres-

[4] See SAE Transactions, Vol. 29, 1934, pp. 444-453: "Engine Lubrication with Different Bearing Metals; Especially Copper-Lead," by C. M. Larson.

[5] See *SAE Journal*, Vol. 15, July, 1924, pp. 5-6: abstract of "Corrosion in Automotive Engines," by H. C. Mougey.

[6] See SAE Transactions, Vol. 19, Part II, 1924, pp. 58-81: "Water in Crankcase Oils," by A. L. Clayden.

[7] See SAE Transactions, Vol. 10, Part II, 1915, pp. 86-109: "Automobile Lubrication," by C. W. Stratford.

[8] See SAE Transactions, Vol. 17, Part II, 1922, pp. 158-199: "Oil Consumption," by A. A. Bull.

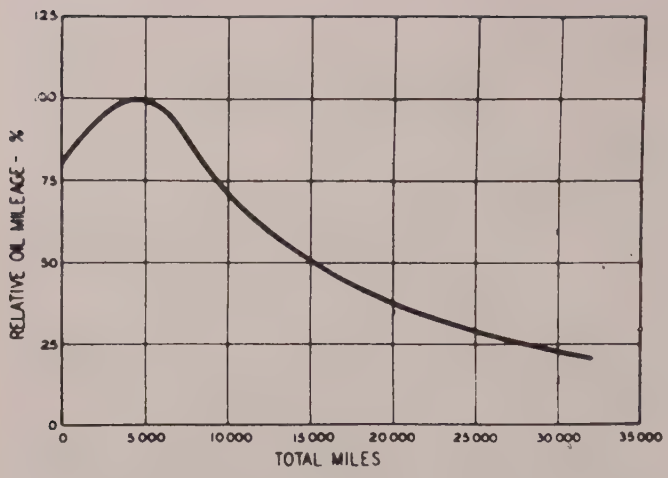

Fig. 2 – Relationship between engine age and oil mileage (from Mulit and Kavanagh)

ent is about 1 gal per 1000 miles (250 miles per quart). Keeping between that figure and 1 gal per 400 to 500 miles (100 to 125 miles per quart) is doing pretty well."

Up to that time oil-control rings of the designs with which we have been familiar for about 25 years were unknown, but by 1925 the slotted oil-control rings were coming into general use. In an advertisement in the January, 1925, issue of *Motor*,[9] a piston-ring manufacturer, in telling of the merits of his new slotted "oil-regulating" ring, says:

"One of these rings installed in the lower groove of each piston . . . makes possible an oil mileage of at least 1000 miles to the gallon."

These oil-consumption figures, representative of these early days, were obtained under the more favorable conditions of operating at car speeds very much less than those that are typical of driving today, and if allowance is made for the much more severe service typical of modern driving at higher speeds, it is apparent that the oil and automobile industries have made enormous improvements in oil consumption.

Many factors must be considered when the subject of oil consumption is studied. Davis and Best[10] showed that oil volatility is just as important a factor as oil viscosity. Mulit and Kavanagh,[11] in discussing oil consumption, point out the effect of several factors, such as the effect of car speed, effects of viscosity and volatility of the oil; and they show that oil consumption changes with the number of miles the car has been driven. This is shown by Fig. 2, which is reproduced from the paper by Mulit and Kavanagh. While definite mileages are given in the figure, Mulit and Kavanagh explain that these are "average" and the rate at which the oil consumption changes with the number of miles of service of the car depends upon how rapidly the piston rings and cylinders wear. In other words, the ability of an engine oil to decrease wear is just as important a property of the oil in its effect on oil consumption as viscosity or volatility. Fig. 3 is from a paper[12] by Kavanagh presented in 1952. He shows how a heavy-duty oil gives a much better oil mileage record in taxicab service than a premium oil, that is, an oil that does not contain the corrosion resistance that is normally imparted by the detergents in a heavy-duty oil. In their 1941 SAE paper, Mulit and Kavanagh say: "The cause of the extremely high wear rate at low operating temperatures is corrosion of the cylinder walls and piston rings by acidic combustion products."

This same story is told by Palmer.[13] Fig. 4 is from a paper by Palmer which was presented in 1952. It shows the same story in cars operated in normal home-to-work service.

This same story is also illustrated by Fig. 5, from a paper presented by Larson[14] in 1953. He shows how a straight mineral oil was responsible for high oil consumption in a taxicab fleet and how, by the use of a Supplement 1 oil, this oil consumption was greatly improved by decreasing the rate of engine wear. Incidentally, the lengthening of the time between engine overhauls, due to the decrease in the rate of wear resulting from using the Supplement 1 oil, is of much greater value to the fleet owner or car owner than the simple matter of the improvement in oil consumption.

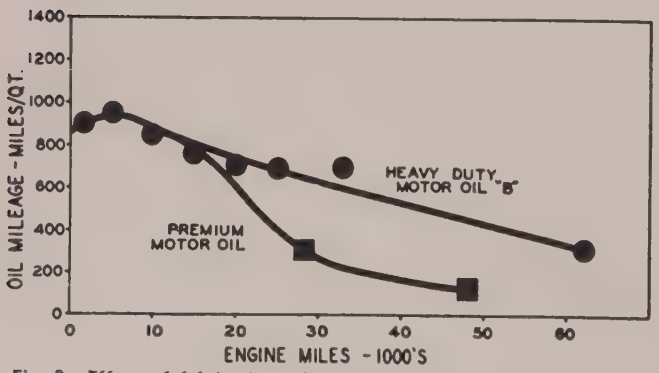

Fig. 3 – Effect of lubricating oil on engine life as shown by rate of change of oil mileage (from Kavanagh)

Break-in Oils and E-P Crankcase Oils

For many years there has been a difference of opinion as to the kind of oil that should be used for break-in and for the original fill in crankcases.

[9] See advertisement on p. 211 of *Motor*, Vol. 42, January, 1925.

[10] See ASTM Proceedings, Vol. 35, Part 2, 1935, pp. 608-626: "Effect of Volatility on Oil Consumption," by L. L. Davis and R. D. Best.

[11] See SAE Transactions, Vol. 48, March, 1941, pp. 98-106: "Lubricating Oils for Internal-Combustion Engines," by L. H. Mulit and F. W. Kavanagh.

[12] See SAE Transactions, Vol. 61, 1953, pp. 231-232, 236: "Engine Wear – Comparison of Radioactive Wear and Field Measurements," by F. W. Kavanagh.

[13] See SAE Transactions, Vol. 61, 1953, pp. 237-243: "Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service," by J. L. Palmer.

[14] "Engine Deposits and Wear," by C. M. Larson. Presented at 18th Midyear Meeting of API, New York City, May 14, 1953.

Some automobile companies believe that the car will be driven at lower speeds during the first few miles and that as soon as the car owner receives his car he will probably change the oil to the kind that he prefers, and that for this reason the kind of oil used for original fill is not very important. Other automobile and oil companies believe that the corrosion-resistant properties of a heavy-duty oil are very important, and, in addition, in many cars, due to the use of piston rings of the newer types, which have higher unit loads than the piston rings used a number of years ago, they believe that the oils should also have extreme-pressure properties. This latter point is illustrated in the following statement:[15]

"A straight mineral oil normally provides fluid lubrication with a high safety factor, however, under special conditions such as rapid break-in, using high speed and power output, a critical loading may be attained to such an extent that fluid and boundary lubrication are inadequate, permitting incipient metal-to-metal contact, leading to ring or cylinder-wall scuffing. Under these conditions the sulfur, phosphorus, or sulfur-phosphorus types of addition agents are beneficial, and the types of these compounds previously discussed in connection with oxidation stability and bearing corrosion protection also contribute a secondary action of increasing 'film strength.' These additives do not form a ferrous sulfide or phosphate film on the ring or cylinder-wall surface at temperatures of normal operation. However, when temperatures rise rapidly at conditions of incipient metal-to-metal contact, a metal sulfide or phosphate film may be formed, and thereby provide an increased factor of protection."

One automobile company has found that in new cars, and especially in cold weather, when the viscosity of the oil in the crankcase may be very high, the load on the gear driving the oil pump and the distributor may be so high that scoring may result if the oil does not have good extreme-pressure properties. They have found that if the gear is well run in with an oil high in extreme-pressure properties the factor of safety in service is greatly increased, and they use this gear test to select the oil to be used for break-in and for original fill.

On the other hand, one automobile company suggests that only oils without additives should be used for break-in or for use during the first part of service. In a June, 1953, publication, which has wide distribution to service managers, parts managers, shop foremen, and mechanics, they say:

"During the early life of a new engine a certain amount of piston-ring and cylinder-bore wear is necessary to seat the rings properly. Unless this normal amount of new-engine bore and ring wear

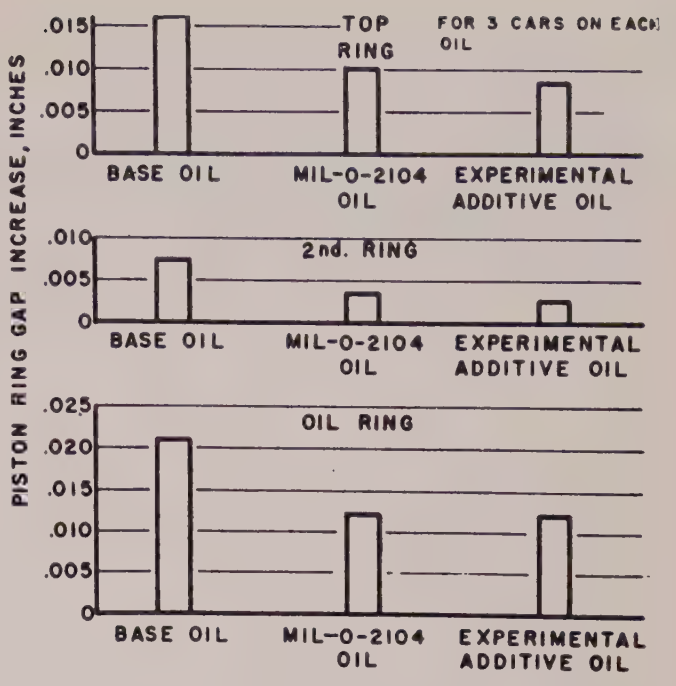

Fig. 4 – Average piston-ring wear as affected by oil type – 1950 Powerglide engine, 236 cu in., 14,000 miles of home-to-work service (from Palmer)

takes place, oil consumption may remain abnormal for several thousands of miles.

"Improper ring seating in a new engine may be due to mild operation or to the use of an oil containing additives intended to reduce wear. In cases of this type it is suggested that oil containing no additives, or with a low additive level, be used in the crankcase. These oil types are usually designated as 'for Service ML and/or MM' only, as described on this page (of the June, 1953, service publication).

"After normal consumption is obtained, a good grade of oil suitable for the service intended should be used."

However, the automobile company making this statement says that it is not intended as a general statement of policy, but it is simply a suggestion

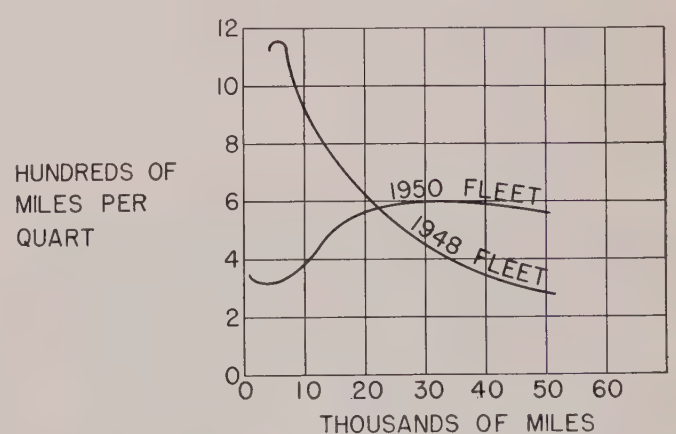

Fig. 5 – Oil consumption of New York City taxicab fleet. 1948 fleet – straight mineral oil; 1950 fleet – supplement 1 oil (from Larson)

[15] See *Lubrication*, Vol. 38, January, 1952, pp. 1-12: "Modern Automotive Engine Oils."

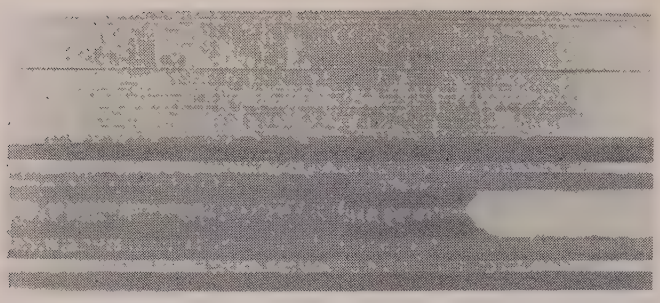

Fig. 6 – Piston rings after typical 100-hr test using commercial heavy-duty SAE 20-20W oil

in a service publication in regard to how the service man may take care of isolated complaints which may occur in exceptional cases. Attention is called to the words in the statement, "In cases of this type," which indicates the limited application of this statement.

The piston-ring manufacturers recognize this problem and they try to take care of it by the design of their piston rings. In the advertisements of the piston-ring manufacturers there are many claims to this effect. For example:

In *Bus Transportation,* August, 1953, p. 3, one piston-ring company says: "All rings are beveled or tapered to thread line contact for quick seating and blowby control."

In *Motor,* July, 1952, p. 71, another piston-ring company claims that its rings give quicker seating and faster oil control without costly engine drag.

In *Motor,* August, 1952, p. 1, a third piston-ring company says: "There's no tedious break-in period with ——— rings – they are factory lapped for quick, positive seal."

In *Motor,* August, 1952, p. 70, a fourth piston-ring company says that a coating is used on its piston rings. This company says, in describing this coating, that it is "a coating of natural bearing material for quick seating and extra protection against scuffing, wear, and acid action."

In *Motor,* June, 1953, p. 1, a fifth piston-ring company says: ". . . chrome piston-ring set . . . seals compression and controls oil for over twice as long as ordinary rings. Seats faster than unplated rings."

Other oil companies also recognize this problem. Snyder, Martin, Raymond, and Socolofsky[16] said recently:

"Special break-in oils developed by various laboratories have been used to advantage by the engine builder to promote faster and safer break-in. However, data available to the authors indicate that, in engines of higher output or operating under more severe conditions, the benefit derived from current oil formulations is small compared to factors within the control of the engine builder . . ."

From this discussion it appears that some believe break-in oils are unnecessary, some believe they are necessary, and some believe it is a problem that should be solved by engine design. However, even though the break-in problem should be solved by some suitable method, and even though in some of the engines that were designed before World War II, and also in more recently designed engines, if the loads and speeds are kept low, it may be difficult to show much advantage in oils that are high in extreme-pressure properties, it is a fact that oils high in extreme-pressure properties are necessary to reduce wear in at least some of the newer high-compression engines, especially if they are operated at high power and high output.

One of the tests which is regularly made by automobile engineers whenever a change in design in an engine is considered is the 100-hr full-load full-speed test. Engineers consider that no change in any detail should be made in an engine until it is established that this change will not be objectionable from the standpoint of durability as shown by this 100-hr test.

Fig. 6 shows the piston rings from a typical 100-hr test of a high-grade commercial heavy-duty SAE 20-20W oil regularly used in one of our car division engineering departments. For convenience, we will call this oil, oil A. Note that the tool marks are still visible on both the top and second compression rings at the end of the 100-hr test. These results are very good. Similar good results are also obtained in other General Motors car divisions using oil A in either the 10W or 20W grades.

Last summer, in making our tests on the new SAE 5W oils, which were introduced last winter, a 100-hr full-load, full-speed test was made on one of the commercial SAE 5W oils, which we will call oil B. Fig. 7 shows the bad ring wear and scuffing obtained in this test with this particular SAE 5W oil. At first, some of our engineers thought that SAE 5W oils, in general, might not be so good in resistance to ring wear and in oil consumption as SAE 10W or SAE 20W oils, but when it was pointed out that the good SAE 5W oils are as high in viscosity at 210 F as the 10W or 20W oils, it was appreciated that the trouble with oil B was probably due to lack of extreme-pressure properties.

To throw more light on this question, three additional 100-hr full-load full-speed tests were run. A commercial SAE 20W oil, which was widely marketed, was selected. This oil will be called oil C. This oil was a well-known heavy-duty oil, high in detergency, but low in extreme-pressure properties. This 20W oil C was found to give as bad ring wear as oil B. As soon as the oil company making oil C learned of these bad wear tests, they raised the extreme-pressure properties of their oil and, as shown by engine tests made by two different

[16] "A Broad Look at Engine-Oil Testing," by G. H. S. Snyder, E. A Martin, L. Raymond, and J. F. Socolofsky. Presented at 18th Midyear Meeting of API, New York City, May 14, 1953.

General Motors divisions, this oil is now very satisfactory in resistance to wear and in extreme-pressure properties. As soon as the oil company making the SAE 5W oil B learned of the bad ring wear in the engine tests, they also increased the extreme-pressure properties of their oil, and the results of a 100-hr full-load full-speed test on this improved SAE 5W oil are shown in Fig. 8. The resistance to wear, as measured by this 100-hr full-load full-speed speed test, is just as good using the revised SAE 5W oil as when good SAE 10W or SAE 20W oils are used. In addition, a number of road tests, including several 25,000-mile road tests conducted under severe operating conditions, have shown that the good oil consumption properties and the good resistance to wear, shown by the dynamometer tests, are also shown in actual road tests with this improved SAE 5W oil.

Problems Relating to Break-in Oils and E-P Oils

In view of these differences of opinion in respect to break-in oils, and in view of the importance of extreme-pressure crankcase oils, I would like to propose some more problems that should be solved by the oil and automotive industries. Experience has shown that the extreme-pressure testing machines developed for testing oils for hypoid-gear rear axles do not correlate with service, and that, consequently, it is necessary to make tests on rear-axle parts under testing conditions simulating extreme operating conditions in actual service. This situation is well known. As long ago as 1885, F. W. Arvine,[17] chemist of the Standard Oil Co., pointed out that the agreement between results yielded by ordinary test machines and those obtained in actual practice was, unfortunately, very small. Of course, after the tests made in the laboratory are completed, it is always necessary for the results to be checked in actual service on the road under the most severe conditions expected to be encountered.

I would like to propose that a test or tests be developed to evaluate extreme-pressure properties in crankcase oils. The 100-hr full-load full-speed engine test is very good and very positive, but since it uses up an engine per test, it is not very good for research work by the oil companies or for inspection tests by the automobile companies for checking incoming shipments. We are working on a device using the actual oil pump and distributor gears and the actual camshafts used in production, and we hope it will be found to be satisfactory for research and inspection work. We are also working on a modification of the L-4 test commonly made on the Chevrolet engine. In this modification, we use a Chevrolet Powerglide engine in order to be able to use the hydraulic valve lifters, and, in our preliminary tests, it appears to have considerable merit as a test for extreme-pressure properties. Perhaps other better tests may soon be developed as other laboratories work on the problem.

After the suitable test machine or test procedure is developed, it will be necessary to establish how high in extreme-pressure properties a crankcase oil should be. This means the results must be correlated with service, especially service under the most severe operating conditions that will be experienced.

Another very important problem is the compatibility of the various extreme-pressure agents with copper-lead bearings, bronze pistons-pin bushings, and so forth. We have developed some bearing test machines that can be operated under very severe operating conditions, and we can show that in these machines, under the very severe test conditions, the copper-lead bearings and/or the bronze piston-pin bushings will be attacked by certain addition agents. In some cases we can also show the same results by some very severe road tests, especially in engines that are not thoroughly broken in, and yet in actual service we have not experienced any trouble with these particular additives. We should not develop tests that are carried to such extremes that they exclude desirable products from use in service.

Some of the piston-ring manufacturers claim that their rings operate at low pressures, even though tests may show that certain other piston rings that operate at high unit pressures may be very desirable in reducing oil consumption. What are the limits in pressure that good extreme-pressure crankcase oils will meet? In addition,

[17] See ASME Transactions, Vol. 6, 1885, pp. 160-164: discussion by F. W. Arvine.

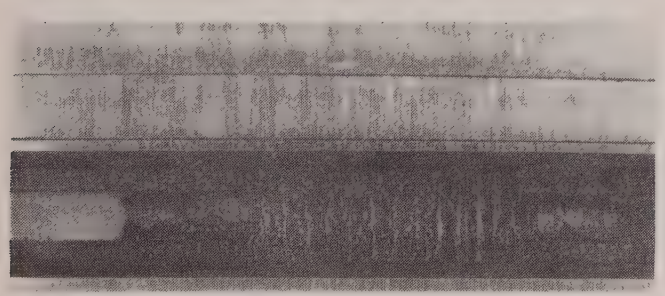

Fig. 7 – Piston rings after 100-hr test with commercial SAE 5W oil

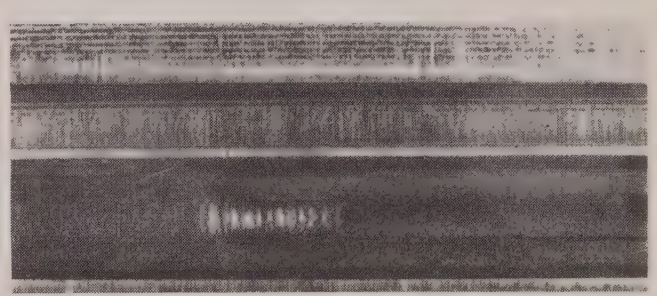

Fig. 8 – Piston rings after 100-hr test with improved SAE 5W oil

chrome-plated compression rings and both steel and chrome-plated oil-control rings are rapidly coming into service. The work on extreme-pressure properties of the crankcase oils should recognize these newer piston-ring materials, and it should be established how high in extreme-pressure properties the crankcase oils should be to take care of these newer piston-ring materials.

Hydraulic Valve Lifters

Hydraulic valve lifters have been used for many years, but when hydraulic valve lifters came into more general use a few years ago, numerous complaints in regard to sticking were obtained. Some people said that the early lifters did not have this trouble, but in the paper actually presented at the SAE Annual Meeting, January, 1939, on "Recent Opinions on Varnish in Engines," by F. F. Kishline, there were some remarks that do not appear when the paper[18] was condensed for publication. Although many engine men reported stuck pistons, rings, valves, tappets, and the like, one engine manufacturer replied:

"... In field operations, the only troublesome item from the varnish formation is sticking of hydraulic lash adjusters. In these parts, the operating clearance between plunger and cylinder is 0.0003 to 0.0004 in. and operating pressure differentials between the units relatively low, so that sticky units caused by varnish deposits have to be removed from the engine and thoroughly cleaned to restore original operating efficiency. The parts became temporarily inoperative but do not fail."

Of course, these statements made in 1939 were before the general use of heavy-duty oils in automobile engines. Now that heavy-duty oils are generally used in cars equipped with hydraulic valve lifters, the problem of lubrication of hydraulic valve lifters is no longer a serious one. Under the more severe operating conditions of stop-and-go service in city operation, especially in winter, the heavy-duty oils that are especially resistant to ferrous corrosion give better results. A number of oil companies are now marketing Supplement 1 heavy-duty oils at filling stations for passenger cars, but until we have had more experience in actual service, no automobile company is willing to say that these Supplement 1 oils are absolutely required and that oils superior to MIL-0-2104 must be used.

Crankcase Oils Designed to Reduce Octane Requirement Increase in Engines

In 1921 C. F. Kettering[19] made the following remarks, in discussing cooperation between the automotive and oil industries:

"What we want in the automotive industry is a fuel and a lubricating oil that will give good efficiency and obviate the necessity of cleaning the carbon out of the engine. I believe that with the right kind of cooperation, that can be attained quickly. Carbon deposits contain a mixture that is nothing but a varnish, and acts in exactly the same manner. If that 5 or 10% of binder is dissolved out, the rest of the carbon goes out, and there is no trouble with carbonization."

There have been many papers on carbon deposits in engines,[20, 21, 22] but special attention should be called to the work of Livingstone and his associates[23, 24] on carbon formation in engines, and to the work of D. P. Barnard and his associates[25] on the "carbonization index." Viscosity index improvers,[26] with all of their wonderful properties, were introduced about 1934, and oxidation inhibitors and detergents followed a few years later. It appears that for many years we have had all the information we really needed, but it was not until the remarkable effect of some of the new SAE 5W oils was noticed that there was enough commercial urge to cause the oil industry to put on the market oils designed to reduce the octane requirement increase in engines.

Problems Relating to Oils to Reduce Octane Requirement Increase

The oil and automotive industries are in great need of tests to evaluate oils in respect to their ability to reduce octane requirement increase. A number of oil and automobile companies now have such tests, both dynamometer tests and road tests, but these tests require the operation of engines or cars for long periods of time. Screening tests, perhaps tests developed on evaporation tests or vacuum distillation tests, are desired for research work and to help in selecting the oils that should be tested in the long-time dynamometer or road tests. The other factors in addition to the volatility characteristics should also be developed, since it is well known that there are other factors that must be considered.

Hypoid-Gear Rear-Axle Lubricants

Hypoid gears have been used in passenger cars in rear axles since 1926. Prior to 1936 they were

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[18] See SAE Transactions, Vol. 34, 1939, pp. 321-324, 334: "Symposium on Varnish in Engines," by F. F. Kishline.

[19] See API Bulletin No. 132, Dec. 10, 1920, pp. 35-41: "Consumption – the Automotive Industry," by C. F. Kettering. Paper also abstracted in *SAE Journal*, Vol. 8, January, 1921, pp. 43-45, under title "Cooperation of Automotive and Oil Industries."

[20] See SAE Transactions, Vol. 21, Part I, 1926, pp. 21-34: "Tests of Carbon Deposition in Internal-Combustion Engines," by D. P. Brooks.

[21] See SAE Transactions, Vol. 22, Part II, 1927, pp. 15-20: "Quantitative Effect of Engine Carbon on Detonation," by N. MacCoull and D. P. Brooks.

[22] See SAE Transactions, Vol. 17, Part II, 1922, pp. 200-221: "Oil Pumping," by G. A. Round.

[23] See SAE Transactions, Vol. 22, Part I, 1927, pp. 69-72: "Carbon Depositing Tendency of Heavier Motor Oils," by C. J. Livingstone, S. P. Marley, and W. A. Gruse.

[24] See SAE Transactions, Vol. 24, 1929, pp. 310-316: "Carbon Deposits with Heavy-Duty Engines," by C. J. Livingstone, E. C. Martin, and S. P. Marley.

[25] See SAE Transactions, Vol. 26, 1931, pp. 373-379: "Factors Controlling Engine-Carbon Formation," by W. H. Bahlke, D. P. Barnard, J. O. Eisinger, and O. FitzSimons.

[26] See Proceedings of API, Vol. 15, 15th Annual Meeting, Dallas, Nov. 12-15, 1934, Section III, pp. 76-89: "Improved Motor Oils," by M. Otto, F. L. Miller, A. J. Blackwood, and G. H. B. Davis.

used by only a limited number of automobile manufacturers. The lubrication of these axles did not present a serious problem to the petroleum industry. Approved lubricants, largely of the lead-soap active-sulfur type, were handled by the authorized car dealers and no attempt was made to market a hypoid lubricant through the filling stations.

In 1936 hypoid gears were adopted for quantity production by a large number of passenger-car manufacturers. The widespread use of hypoid-gear axles made it necessary to expand the sources of supply for hypoid lubricants quickly. The success of the whole program, and the almost complete absence of service failures or complaints were due to:

1. The extensive educational campaign carried out by both the automotive and petroleum industries.

2. The cooperation of the petroleum industry in marketing a powerful extreme-pressure or hypoid lubricant through all retail lubrication outlets.

Lead-soap active-sulfur lubricants were used successfully in all makes of passenger-car hypoid-gear axles. However, when hypoid gears were introduced into truck axles about 1939, it was found that the lead-soap active-sulfur lubricants were too active chemically for the truck axles, and a lubricant that was less active had to be developed for use in the truck axles. Unfortunately, most of the less active truck lubricants were too weak for use in passenger-car axles, and consequently, for a short period of time, it was necessary for the petroleum industry to carry two lubricants, a powerful lubricant for passenger cars and a milder one for trucks.[27]

To avoid the mistakes due to getting these lubricants confused and used for the wrong service, the petroleum industry soon developed a single multipurpose lubricant. This lubricant was powerful enough for passenger cars but not too active chemically for use on trucks, and so the single multipurpose lubricant soon came to replace the two lubricants previously supplied at filling stations, even though most of the car manufacturers continued to use lead-soap active-sulfur lubricant for original fill in the passenger-car axles.

Recently, due to the higher loads and speeds of modern cars, it has been found that the SAE 80 multipurpose hypoid lubricants, as a class, are too weak to provide the necessary factor of safety in passenger cars. However, the situation is not so critical with the SAE 90 multipurpose lubricants because the SAE 90 multipurpose gear lubricants, in general, are more powerful than the SAE 80 multipurpose lubricants.

The obvious answer to this problem is the development of a new type of multipurpose hypoid-gear lubricant, more powerful than the present commercial multipurpose lubricant, but still low enough in chemical activity so that it will be satisfactory for use in truck axles. The additive manufacturers and the oil companies are working on this problem, and, at least experimentally, it appears that the problem has been solved. Road tests are now being made to check the laboratory and dynamometer rear-axle tests.

Problems Relating to Hypoid-Gear Lubricants

The first problem is to check the new lubricants on the road under the most severe operating conditions in both passenger cars and trucks to establish that the lubricants will be as satisfactory in service as the dynamometer rear-axle tests indicate, and that they will be equally satisfactory in both passenger-car axles and in truck axles. This work is now in progress. The next problem is the commercial one of developing a new name to distinguish these new lubricants from the multipurpose lubricants now marketed, so that the new lubricants can replace the present multipurpose lubricants in the field and in all recommendations made by the passenger-car and truck manufacturers.

Other Subjects on Which Progress Has Been Made

It is impossible to consider here all of the items on which progress has been made during the last 40 years, or even to tabulate them. Many men have spent their lives working on these problems. Your Committee feels that recognition should be given to some of these men, especially some of those who have spent a quarter of a century or more in these fields of endeavor, and more especially to some who, like myself, are retiring, or who are about to retire. Some of the subjects which should be considered are shown below. In the discussion which will follow this paper, some of these subjects will be covered:

1. Cold starting.
2. Pour points, pour-point depressors, stable pour points, and pumpability at low temperatures.
3. Solvent refining.
4. Viscosity index and viscosity index improvers.
5. Oxidation inhibitors, bearing corrosion, and sludge and varnish.
6. Diesel-engine oils, detergents, and other additives.
7. Oil classifications, such as the API system.
8. Oil filters and air cleaners.
9. The new SAE 5W-20 crankcase oils and the new SAE 10W-30 oils.
10. Automatic transmission lubricants.
11. Chassis lubricants and other greases.

The list of important subjects could be extended to cover very many more problems, but there is still one subject that is more important than all of the others combined. It is the problem of cooperation of the automotive and oil industries. This was the theme of the address given by C. F. Kettering at

[27] See *Oil and Gas Journal*, Vol. 38, Nov. 17, 1939, pp. 162, 164, 167, 168, 170, 171: "Hypoid-Lubricant Requirements for Passenger-Car and Truck Rear Axles," by H. R. Wolf.

the first annual meeting of the American Petroleum Institute in 1920. Reference to this paper has already been made.[19] Mr. Kettering said:

"We have to carry this movement on together as a coordination between the American Petroleum Institute, the automotive industry, and the chemical societies, because it is possible, and we know enough today in a good many ways to help this situation out."

At first this cooperation was rather limited, but a number of years ago its importance was recognized by L. C. Welch, who, for a number of years, was chairman of the API Lubrication Committee. Through his efforts steps were taken to develop this cooperation, and this policy has been followed by the men who took his place after he retired.

The Coordinating Research Council, sponsored jointly by the SAE and the API, is a very important factor in this work of cooperation. The SAE Fuels and Lubricants Technical Committee is another very important factor. Interlocking membership on committees of the SAE and the other national societies is another important help. The SAE itself, of course, is the society which is composed of the two industries, and the officers of the SAE are continually doing everything they can to further this cooperation.

I want to second the plea[28] made by Joe Moller at the Annual Meeting in 1938 when he presented a paper on "What is an E-P Product?"

"Mr. Moller closed with a plea for joint cooperation between suppliers and users, so that problems will be solved before, rather than after, they become urgent commercially."

I think we can all be proud of the progress we have made in the last "forty years," and as a result of this cooperation which we have between our two great industries, I am sure that the progress we will make in the next "forty years" will continue to the great benefit of the oil and automotive industries, and to the benefit of our mutual customer, the public.

[28] See *SAE Journal*, Vol. 42, February, 1938, p. 36: digest of "What is an E-P Product?" by J. Moller.

DISCUSSION

"It Costs Too Much" Is a Poor Excuse

– Frank A. Suess
Continental Oil Co.

MR. Mougey has discussed many important subjects, but I will mention briefly only a few of them.

As far as crankcase dilution and crankcase corrosion and their relationship to wear are concerned, certainly the following three conclusions are important and useful:

1. Corrosion from water and acidulous water cause wear.
2. Proper selection of fuel and keeping operating temperatures high enough to prevent condensation of the water and acids decreases the corrosion and resulting wear.
3. Crankcase oils with the ability to "rustproof" the engine parts will decrease the corrosion and resulting wear.

Is it unreasonable to wish that some attack be made on the cause of the trouble, or that really effective crankcase ventilation be employed to remove the greater portion of the condensate and acids before they do damage? Thus, it seems to me, a fourth conclusion should be that effective crankcase ventilation is needed – especially at low car speeds. We know how to do it, but 'tis said, "it costs too much money." So what? How much? Maybe a couple of dollars at the most for something that is useful and definitely beneficial. Cost does not seem a deterrent to such things as a trick mirror on the sun visor for the gals to look into, or false wire wheels to get dirty, or air-conditioning, and-so-forth. Nor is there much reluctance to charge a suitable price for them. Detergents are beneficial relative to corrosion but can be even more so when excessive moisture is avoided rather than tolerated.

Concerning factors affecting oil consumption, Mr. Mougey points out that "Davis and Best showed that oil volatility is just as important a factor as oil viscosity" and with the new SAE 5W-20 or 10W-30 oils available today the volatility factor becomes extremely important and worthy of study by every refiner producing these SAE viscosity grades.

On break-in oils and extreme-pressure crankcase oils, Mr. Mougey lists a very comprehensive outline of seemingly well-organized confusion. A little like the popular song "Undecided" – "it's undecided now – so what are we going to do?" Some believe break-in oils unnecessary, others contend they are desired, and others believe it is a problem that should be solved by engine design. Certainly, the latter merits considerable thought and action. And I certainly have contended for over 20 years that improved load-carrying properties, film strength, or – as described in this paper – extreme-pressure properties are desirable. I, for one, dislike the term extreme-pressure applied to a motor oil for I seriously question if we really know that the property desired is extreme pressure. Furthermore, I urge that we make haste slowly in the matter of devising some test that will evaluate these properties in oils. Better that we first try to find out what we are trying to measure. Otherwise, we – both the automotive and petroleum industries – may foster a test procedure that will later rise up to haunt us and cause the unnecessary expenditure of great sums of money.

For example, right today a 100-hr full-load full-speed test is used to find an oil property that will let certain newer high-power engine parts live when operated under these extremes of conditions which but seldom occur in service – while other newly designed high-speed, high-powered engines have parts that dislike the heavily fortified or high-additive oils. Thus, we must see – and should certainly admit – that we do have problems – design problems and metallurgical problems just as much as oil problems. Certainly we can and will solve them but let us hope and strive for an orderly cooperative effort in the next "Forty Years" between these two industries in serving our mutual customer and avoid, as far as possible, any hasty or panicky expedients.

Standardization for Automatic Transmission Fluids

– H. R. Wolf
GM Research Laboratories Division

MY discussion will be limited to automatic transmission fluids.

The introduction by Oldsmobile of the Hydra-matic drive, as optional equipment in 1940 and by Cadillac in 1941, presented a new problem to the petroleum and automotive industries. Previous service experience, over a period of approximately two and one-half years, with a semi-automatic transmission, demonstrated very conclusively that fluids having the required performance characteristics and capable of functioning as an integral part of a fully automatic transmission were not available to car owners through the petroleum industries retail marketing channels.

The original version of the now well-known type A fluid was developed by the GM Research Laboratories Division prior to 1940. Due to slight mechanical differences in the early models of Hydra-matic transmissions, two different fluids were required for original fill and a third modification was required for service in one of these transmissions. To complicate the problem further and to prevent standardization, other equipment manufacturers were developing other types of automatic transmissions that required different fluids.

Some 10 years later, in 1949, standardization had progressed to the point where it was possible to establish the Armour Research Foundation program for qualifying fluids and to invite the petroleum industry to participate in the distribution of automatic transmission fluid. At the present time, a large number of major refiners are manufacturing fluids of their own formulation, and approximately two hundred individual authorized brand-name fluids, identified by Armour Qualification Numbers, are available to car owners and car dealers.

Incidentally, it is gratifying to be able to report as one milestone in this "Forty Years of Progress in Automotive Lubrication" that all manufacturers of fully automatic transmissions are now recommending the use of the type A fluid.

The extensive work of the SAE Subcommittee on Fluids for Fluid Couplings and Torque Converters, the CLR Power Transmission Fluids Group, and the ASTM D-2 Section on Hydraulic Power Transmission Liquids should be acknowledged. These groups have been extremely helpful in disseminating information regarding the properties and performance characteristics of automatic transmission fluids.

Cooperative work by the petroleum and automotive industries over the past 40 years has resulted in the development of improved products and in the reduction of the number of viscosity grades and types required for service applications. New problems will undoubtedly result from the ever-increasing use of automatic transmissions. These problems, as they arise, will be solved through the continued cooperative efforts of the two industries.

Solving the Carbon Deposit Problem

– C. J. Livingstone
Gulf Oil Corp.

MR. Mougey has been kind enough to mention in his paper some of the early work with which I was associated at the Mellon Institute back in the 1920's. In those days carbon deposit was a very serious problem, since it was not unusual for car owners to remove the cylinder heads every 500 to 1000 miles for cleaning carbon. It was in connection with this work that I first met Mr. Mougey and perhaps the circumstances of our meeting have been repeated many times with others as he enlisted their aid in solving his problems.

In 1927 I was coauthor with S. P. Marley and W. A. Gruse of a paper on the subject of carbon deposits, presented at the SAE Summer Meeting at French Lick. Immediately after the meeting, Mr. Mougey came to us and told us of a serious carbon deposit problem occurring in sleeve-valve engines manufactured by a division of his company. Since we had shown a considerable difference between lubricating oils in laboratory engines, he proposed that we take part in a cooperative fleet test using buses powered with sleeve-valve engines. Our laboratory work had indicated that there was a relationship between the Conradson carbon residue of a lubricating oil and its carbon deposit tendency. Consequently, he suggested that an oil of natural low carbon residue, one with a high carbon residue, and a third oil in which the carbon residue had been drastically reduced through a new refining process, be included in the tests. The fleet test developed that there was a wide difference in the performance of the lubricating oils in these engines, but the relationship between carbon residue and carbon deposit tendency was not confirmed. Oil volatility and the characteristics of the deposit formed appeared to be the most important factors.

When the fleet test was completed, the question of publication of results came up. An executive of the division of Mr. Mougey's company which was involved was greatly concerned regarding the commercial implication of our engine performance data. Mr. Mougey championed the right and duty of technical people to publish their results, regardless of the effect it might have on the commercial acceptance of sleeve-valve engines. I was greatly impressed by the technical integrity of Mr. Mougey at this time, as I have been many times over the years I have known him.

Oil Industry Efforts To Improve Oil Quality

– D. P. Barnard
Standard Oil Co. (Ind.)

"CLEANLINESS is next to Godliness." Mr. Mougey's work in the field of motor oil development can, with certain exceptions, be said to have been dominated by this ancient adage. In the very beginning he recognized the importance of contamination – as, for example, by acidified and therefore highly corrosive, condensed moisture. Engines got dirty. Combustion-chamber "carbon" made them knock and sludge befouled rings and oil passages. As the Mougey references point out, Brooks in 1926 reported tests on carbon formation; later, Livingstone, Marley, and Gruse showed that certain oil characteristics were at fault. In 1931, Bahlke et al suggested a volatility test that could be used to measure carbon-forming propensities.

In those earliest days, color stability, pour point, and the debatable carbon residue tests were about all that were available to describe motor oil "quality," which depended, to a large extent, on crude sources. Coastal crudes provided low pour point and low carbon-forming oils; Pennsylvania crudes provided low consumption, easy cranking, and highly stable oils; while Midcontinent crudes neither excelled in these properties nor were they too deficient. Prior to the advent of solvent refining, no single crude would yield a motor oil meeting the desirable standards for low carbon-forming tendency, low pour point, high viscosity index, and resistance to oxidation. In developing solvent refining and solvent dewaxing, the oil industry took the most significant steps in the history of motor oil evolution. They form the basis for our present high V.I., highly stable, low pour point, non-carbon-forming motor oils, which can cope with virtually every service requirement – however extreme.

The first large-scale solvent extraction of lubricating oils used SO_2, but while this solvent was considered best for re-

fining kerosene from coastal crude it failed to produce motor oils that would even approach the viscosity index characteristics of Pennsylvania oils. The first commercial extraction process to make motor oil of the desired high viscosity index from a low V.I. stock was the Phenol plant installed by Imperial Oil Co., Ltd., in Sarnia, Ontario, Canada in 1929. Standard of Indiana built its first Chlorex unit in 1932, and shortly thereafter introduced its first improved high V.I. "antisludge" motor oil. This product, with its inherent low carbon-forming tendency and other desired properties, set the pattern for basic motor oils that generally prevails today. Other processes, notably Furfural, Duosol, Nitrobenzene, and SO_2-Benzol followed in quick succession, until now the total U. S. solvent-refined lube oil capacity is over 160,000 bpd of finished oil. This is equivalent to about 90% of the total U. S. lube oil production. On a charging stock basis, U. S. capacity alone amounts to about 200,000 bpd for solvent extraction, 132,000 bpd for solvent dewaxing, and 60,000 bpd for propane deasphalting. Propane removes asphaltic components either prior to or during solvent extraction.

The use of extracted stocks was accompanied by many advantages and also by new and peculiar problems – notably rapid oxygen absorption and the development of organic acidity with consequent corrosion of cadmium and, later, lead bearings. This problem has long since been adequately solved through the use of corrosion inhibitors, which enable the basic merits of solvent-extracted oils to be realized. High V.I., low-combustion-chamber deposition, and good overall stability render doubly effective the various additives for further improving V.I., stability, detergency, and even wear reduction. It is particularly interesting to note that the original virtue of solvent extraction – that it made good V.I. distillate oils in the higher viscosity grades – is now coming in for renewed consideration in the light of the rapidly increasing octane requirements.

Solvent extraction is but one of a number of specialized tools now employed by the lubricant manufacturer. In combination with the judicious use of additives it has made possible advances in lubricants other than motor and engine oils, although it is this last field that depends most heavily on these tools.

Mr. Mougey's long campaign to improve motor vehicle engine lubrication has contributed importantly to the demand for the high V.I., stable, low carbon-forming oils, which play their roles in making possible the high-performance, efficient, and phenomenally durable automobile engine of today.

Manufacturers Must Solve Problems, Too

– J. P. Stewart and G. A. Round
Socony-Vacuum Oil Co., Inc.

MR. Mougey has reviewed in considerable detail past progress made by the petroleum industry in developing better lubricants. He has also mentioned, all too briefly, some of the things the manufacturers have done to improve lubrication performance. By contrast, however, in telling the oil men what problems he thinks they should solve, he omits completely any mention of the things the manufacturers might do, either to eliminate any of the problems he lists or to make them less serious.

The inference is that all of them should be solved through the lubricating oil.

If we look at these problems from the standpoint of our mutual customer, the car owner, it seems to us that he will get far better service at lower cost if both industries accept responsibility for solving the problems Mr. Mougey has listed.

For example, chrome-plated rings reduce ring and cylinder wear and scuffing to a marked degree. For the very small capital cost increase that the owner pays, he saves many times over in oil economy and longer engine life. The improvement is far greater than can be achieved through oil alone.

We might consider the problem of wear of rocker arms, cams, valve lifter faces, oil pump drive gears, and similar parts. One manufacturer has overcome rocker wear while still maintaining effective oil control at the valve stems by a minor change in oil groove design. As a result, hasn't he made a better engine and eliminated a potential service expense? The capital cost of the change in tooling required is negligible compared with the lifetime cost of solving the problem through oil.

Mr. Mougey suggests that extreme-pressure qualities are the answer to those problems, but we know today that greater extreme-pressure or antiweld effect has aggravated valve lifter failures in some cases. In fact, oils low rather than high in e-p rating, judged by the usual yardsticks, have done an excellent job in certain critical engines. We, therefore, think it is not wise to rush into a program of developing a new e-p test machine until we know more about the problem. It may well be that the same amount of time and money could be spent much more profitably in work on a design solution of these problems in the troublesome engines. Let us not forget the SAE machine experience.

Work on gear lubrication has taught us the importance of structural rigidity to assure correct alignment, also of metallurgy and surface finish. Is not the fact that tappets have a spherical face and point contact evidence that rigidity and correct alignment are lacking?

One manufacturer tells us he has eliminated his trouble by improving the surface finish of cams. Another reports that a very critical study of the metallurgical structure of certain parts and more careful control of their heat-treatment eliminated their trouble. Certainly, these are better engines because they are less critical; in the long run they should be less costly to the manufacturer and the car owner.

Since these engine troubles, which Mr. Mougey lists, represent an industry-wide problem in the new designs, isn't it time for them to be recognized as such?

Perhaps the automotive industry, including the component parts manufacturers, would profit by some cooperative activity or exchange of information through the CRC, the SAE, or other cooperative group, in the same manner as Mr. Mougey has proposed so frequently for engine oil developments.

The mere pooling of the available information should be most helpful. In such an effort, the petroleum industry can be counted upon to play its part.

We would also like to discuss Mr. Mougey's suggestions concerning hypoid gear lubrication problems. Several independent surveys have shown that field failures are common with the most potent factory-fill material and that many of these start at the bearings. We see no justification for requiring 100% perfect performance from multipurpose gear lubricants under extreme test conditions when that cannot be achieved in service with factory-approved material.

Test experience indicates that, in some instances, with gears properly adjusted for light load contact, the contact area moves partly off the tooth under test loads. Is this not an indication of inadequate rigidity for current engine power, to say nothing of future requirements? Obviously, there is a wide-open opportunity for axle improvements that will make gears less critical, reduce service expense to the manufacturer, and increase customer satisfaction.

In calling on the oil industry alone to solve these problems, the automotive industry should realize that increasing the demands on lubricants inevitably leads to compromises in performance properties and sometimes imposes limitations on product uses. For example, the extreme antiweld properties of factory-fill passenger-car hypoid lubricants preclude their use for any other purpose.

Getting the Engine To Tell What It Wants

– Harry F. Bryan
International Harvester Co.

IN looking back over the 40-year period with its progress it seems to me that one of the most outstanding lessons learned has been to let the engine tell us what it wants. This has resulted in a number of standardized engine tests, which have gained general acceptance and have proved that reproducible and valid results can be obtained that have industry-wide application.

I remember many discussions in our company as to the propriety of accepting lubricants for our engines which had been qualified on the engine of a competitor. I am sorry to say this same argument has been brought forward in some parts of both industries today. In our experience the results from these test engines are consistent enough to be directly comparable to results on our engine, provided we know and recognize the idiosyncrasies of both the test engine and our own engines.

I am heartily in accord with Mr. Mougey's proposal to develop a suitable engine test to evaluate ferrous corrosion-resistant qualities. The necessity for a test of this type arises from our change in driving and operating habits, stop and go, wide ranges of operating temperature, continuous light-load operation standby, and the like are now the rule rather than the exception.

His discussion of the problems of run-in and preparation of the new engine for its work is certainly apropos when one considers the pressure brought to bear on the engineering departments by the manufacturing departments for shorter run-in periods and fewer operations on the engine between the time it leaves the end of the assembly line and reaches the user. During this period we want the engine to wear in fast and from then on never to wear out.

The progress made in the development of year 'round crankcase oils has been phenomenal, but here again we need and are getting the closest kind of cooperation between the oil and engine industries.

Mr. Mougey again brings up the controversial subject of oiliness versus extreme-pressure qualities of lubricating oils and he believes we need both. There are many points in the modern engine where high film strength in oils is required, such as valve tappets, valve rockers, cams, and oil pump drives, especially if light viscosity oils are to be used. Also, there are numerous points where boundary lubrication is encountered at some time during the operation of the engine.

In our experience the lighter viscosity oils are the worst offenders. My own personal opinion is that we are operating with such thin oil films with these oils that the normal roughness of the surface of even precision bearings, or abrasives normally in the oil penetrate the film and local metal-to-metal contact results.

The reduction in oil film thickness can be due to either high loading, high temperatures, low viscosity, or contaminants – or a combination of all. It is obvious that the engine builder can do a great deal to reduce wear due to conditions of this kind. In our experience the adoption of full-flow filtration has given greater protection. In another case copious lubrication of the trouble point reduced the wear.

However, I believe that in order properly to cover the wide range of requirements due to engine design and maintenance procedures it may be desirable to review the whole subject of oiliness and extreme-pressure qualities in oils in view of the greater demands now being made on crankcase lubricants. Certainly a test or tests should be developed to stimulate the requirements of present-day engines so that oiliness and extreme-pressure properties in the oils can be evaluated.

In closing, I am in complete agreement with Mr. Mougey that the most important problem confronting us is continued cooperation of the automotive and oil industries.

Development of Diesel-Engine Oils

– C. G. A. Rosen and L. A. Blanc
Caterpillar Tractor Co.

THE story of the development of diesel-engine oils, starting with the use of the earliest detergent-type additives in crankcase oils and culminating with the most recent Series 2 oils, is recorded in the technical literature. Details are not necessary here. The aspect of the work that is not so well documented is in fact the area in which our cooperation with Mr. Mougey has been most extensive.

It was recognized early in the development of additive-type oils that the engine owner stood to gain most from these oils if they could be used in a wide variety of engines and services. The earliest detergent-additive oils were, of course, not suitable for many engines because of bearing corrosion problems. By 1939, the first so-called noncorrosive detergent-type oils had been developed and by 1942 it was possible for the U. S. Army Ordnance Department to write a specification for a single type of crankcase oil for all ground force engines. In this specification were included no less than five engine tests. These tests had all been used for some time, two of them having been developed in General Motors laboratories and three in our own.

As the years have increased our field experience with oils developed in this manner, we have been able to eliminate three of these five tests. This has been the result, to a great extent, of our work on the Ordnance Reviewing Committee during these years. Old problems have been solved and new problems recognized. Undoubtedly new tests are needed. But the fact remains that two of the original tests are still substantially the same as they were in 1942. We have not been very successful in our attempt to keep required heavy-duty oils to a minimum number. Today we have four types where we used to have one. It is probable that two would be quite enough.

SAE Crankcase Oil Classification

– C. M. Larson
Sinclair Refining Co.

MR. Mougey refers to the SAE crankcase viscosity number classification being adopted July, 1926, and the SAE 10W and SAE 20W being adopted in October, 1950. He was kind enough not to mention the semi-annual meeting of the SAE at Saranac Inn, New York, June, 1934, where his motion to adopt these 10W and 20W oils was voted down. Thereafter, these oils were carried as Automotive Manufacturers' viscosity classification until 1950. A similar history of patience on his part prevails in the SAE 5W and 5W-20 motor oil developments. Whether the SAE 10W-30 will predominate in the United States will take shape in the near future.

Mr. Mougey's contribution to the cause of service testing motor oils is also outstanding and has done much to put forth this type of qualification where physical and chemical limit specifications prevailed some 10 to 15 years ago. The new API Service Classification for designation of motor oils is the end result of this sound thinking. The same is true of automotive gear lubricants. However, as he states, these API type designations and their respective applications should be overhauled based on present and future trends.

HIGH-TEMPERATURE

IN all metal-working fields, the ability to join metals to metals and metals to nonmetals is of paramount importance. Processes practiced in industry can be divided into two familiar categories: namely, mechanical joining and metallurgical joining. The many variations of welding, forging, brazing, and soldering are all of the family of metallurgical joining methods.

This paper is confined to a particular phase of brazing in which elevated temperature and physical and chemical properties of the brazed part are of prime concern. The intent is to survey the current status of this field and so inform and stimulate potential users of brazed parts. An objective presentation is attempted for a number of commercially available high-temperature corrosion-resistant brazing alloys. Photographs showing successful applications of each are used liberally. Acknowledgment of both brazing-alloy manufacturer as well as part fabricator will be given where possible.

High-strength joints comparable to the base metal itself can be achieved by various welding techniques. By careful design considerations, brazed joints with similar physical properties may also be realized. Whereas sound welds generally exhibit the properties of the base metal, brazed joints by their very nature of application usually are limited to considerably lower operating temperatures. Certain rapidly diffusing brazing alloys,

COMMERCIALLY available high-temperature brazing alloys, with examples of successful applications for each, are described here.

Brazing alloys included are: Nicrobraz, Coast metals alloys, Mond Nickel alloys, Handy & Harman alloys, and Solabraze. Production brazed parts are shown, and details of brazed joints are explained with the aid of photomicrographs.

Developments of new high-temperature sheet-metal assemblies made possible by brazing techniques are discussed. Particular emphasis is given fabrication of high strength/weight ratio structures.

An encouraging future for corrosion-resistant high-temperature brazing is predicted.

The Authors

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BRAZING APPLICATIONS

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This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 1, 1953.

as we shall see, provide interesting exceptions to this generalization.[1]

From an oxidation-resistant and chemical-corrosion standpoint, the choice of suitable braze materials is sharply limited as compared, for instance, to welding of a wide variety of suitable base metals. As is well known, the welding process also has its limitations, and in these areas brazing is frequently the answer.

The field of high-temperature brazing has increased remarkably. Intelligent evaluation of the particular problem, careful production engineering, and controlled brazing procedures have resulted in many noteworthy, commercially successful brazing applications.

Many very significant developments in the brazing field are shrouded by security. This obviously limits the amount of fabrication data that can be released, identification of the part itself, or assembly in which the part is used. Too frequently the limitation is complete, and nothing whatever concerning the particular brazing application can be revealed in the public press.

The brief case histories to be presented have been assembled under these conditions. Therefore, it is hoped that the brevity, and in some cases the anonymity, will be understood.

Solar Aircraft Co. is a major manufacturer of "hot parts" for aircraft and industrial use. Brazing processes are practiced on a large scale in its plants, particularly silver brazing. The broadening but sparsely exploited field of high-temperature corrosion-resistant brazing has been under active development by Solar and other fabricators for a number of years. However, it was not until the advent of jet propulsion that the potentials for high-temperature brazing as a production tool were realized. In the past eight years, remarkable strides have been made both in the United States and abroad. This rapid exploitation is largely the result of military requirements for aircraft and missile production.

Brazing Defined – Let us briefly define the broader field of brazing and then narrow our sights onto the heart of our problem – high-temperature brazing.

Brazing is a thermal process for effecting atomic bonding of solid surfaces by means of an intermediate, fused filler metal.

High-temperature brazing is a metallurgical joining process in which a molten filler metal is used having a melting point in excess of 1600 F but below that of the base metal. Diffusion between filler metal and base metal must occur.

The field of high-temperature brazing encompasses a number of rather specialized applications which deserve mention but not detailed attention.

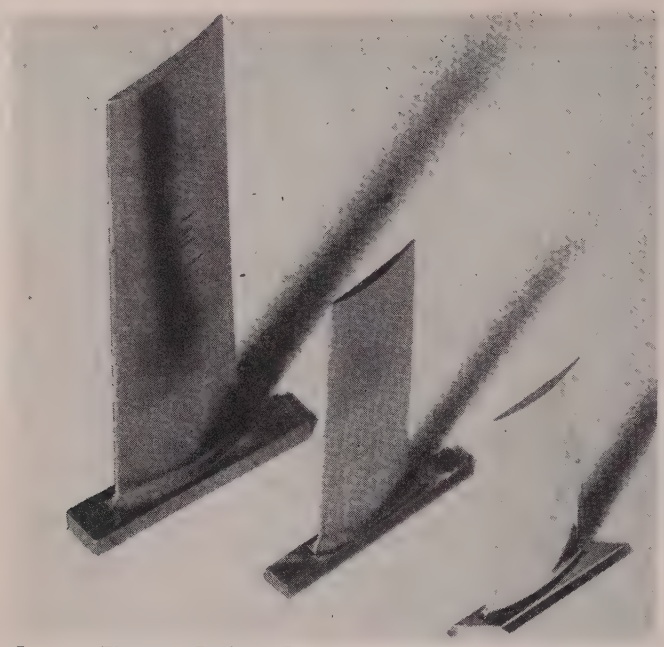
Among the highest-temperature brazes, the elements tantalum, molybdenum, chromium, nickel, and their alloys are used to join tungsten to other refractory metals. In the electric light and electronic tube industry, such brazes are common. In joining molybdenum to tungsten and molybdenum to molybdenum, nickel has been used for years as a braze metal. More recently, a special alloy known as "Mo-braze"[2] (melting point 3450 F) has been introduced.

In the jewelry and dental industries, high-temperature brazes of the platinum group metals are in wide use. Gold alloys should also be included here.

The bonding of metals to nonmetals represents another specialized field for high-temperature braz-

[1] See *The Welding Journal*, Vol. 31, No. 8, August, 1952, pp. 651-662: "Design Properties of Brazed Joints for High-Temperature Applications," by Robert L. Peaslee and Willard M. Boam.

[2] "Mo-braze," American Electro Metal Corp., Yonkers Ave., Yonkers, N. Y.



Courtesy, Thompson Products, Inc.

Fig. 1 – Infiltration-brazing technique produces high-density compressor blading for turbojet engines

ing. In the fabrication of silicon carbide resistor-type heating elements, molten silicon performs a unique braze joining function. The proprietary materials known as "Durhy"[3] in one form consists of carbon particles strongly bonded by means of molten silicon braze metal. Bondley[4] and Pearsall[5] have discovered unusual brazing alloys and techniques for joining of many metals to refractory materials such as glass, fused silica, steatite, and even diamond. Different refractory materials may similarly be bonded to each other.

The powder metallurgist may rightfully be proud of his exploitations of and contributions to high-temperature brazing. Many examples could be cited, particularly if one accepts the infiltration process as a special brazing technique. A truly outstanding development is the commercial production of compressor blading for turbojet engines.[6] Formed porous iron powder compacts are infiltrated or brazed with a molten copper alloy. The infiltrate completely bonds the iron skeleton by capillary action and so produces a high-density blade, as shown in Fig. 1. Minor but significant alloying between iron powder and infiltrate account for the extraordinary strengths observed in this composite material. Additional improvement in physical properties may be realized through metallurgical heat-treatments.[7]

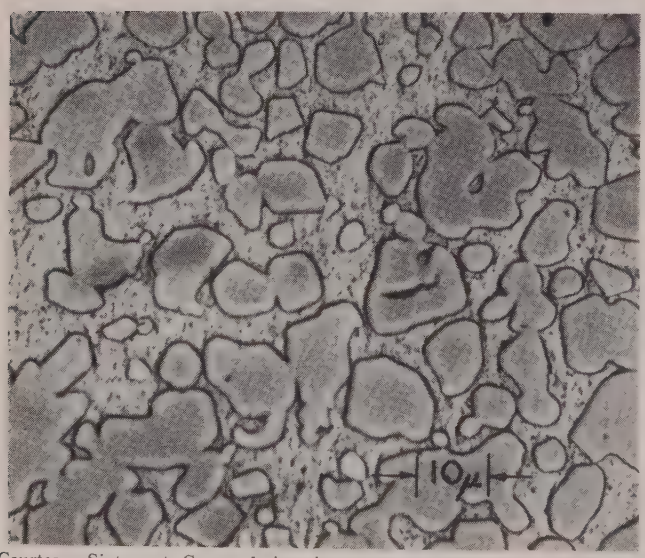
Finally, an extension of this infiltration-brazing technique has profound significance as applied to very-high-temperature gas-turbine blading materials. Again infiltration and bonding of a porous skeleton is accomplished. The porous skeleton in this case, however, is super refractory titanium carbide (melting point above 5400 F); the infiltrate may be any of several super alloys. "Vitallium," "Hastelloy C," "Nichrome V," "Inconel," and other alloys have been employed with notable success as molten infiltrate to effect consolidation.[8, 9] Fig. 2 shows a typical photomicrograph of TC-66-I material at 1000 magnifications.[9] The refractory titanium carbide particles are clearly seen to be further bonded or braze-joined by the continuous super alloy phase.

These high-temperature brazing illustrations considered are of great significance in their specialized applications.

The titanium carbide cermet or metal-ceramic type composite body in particular offers real promise as high-strength, ultra-high-temperature, lightweight structural material.

Emphasis is placed upon high-temperature corrosion-resistant brazing materials. Applications involving braze joining of stainless steel and nonferrous high-alloy parts are presented. Discussion is confined to wrought and cast materials and brazed assemblies made therefrom.

The commercially available brazing alloys to be discussed are: copper, Coast metals, silver-manga-



Courtesy, Sintercast Corp. of America

Fig. 2 – Photomicrograph of TC-66-I material at 1000 magnifications

[3] "Durhy," proprietary compositions developed by The Carborundum Co., Niagara Falls, N. Y.

[4] See General Electric Research Laboratory Research Report, April, 1947, "High-Temperature Metal-Ceramic Brazed Seals," by R. J. Bondley.

[5] See *Materials and Methods*, Vol. 30, No. 1, July, 1949, pp. 61-62: "New Brazing Method for Joining Nonmetallic Materials to Metals," by C. S. Pearsall.

[6] See "TP-1 Sintered Blades for Jet Engines," brochure from Thompson Products, Inc., Cleveland, Ohio.

[7] U. S. Patent 2,456,779, Dec. 21, 1948, "Composite Material and Shaped Bodies Therefrom," issued to Claus G. Goetzel.

[8] See NACA Research Memorandum E52E27, July 23, 1952, "Infiltration of Titanium Carbide with Several Metals," by R. S. Gurnick and A. L. Cooper.

[9] See "Sintercast Titanium Carbide," special brochure from Sintercast Corp. of America, 134 Woodworth Ave., Yonkers, N. Y.

nese, manganese-nickel, palladium-silver, palladium-nickel, Solabraze, and Nicrobraz.

Copper is included for comparative purposes only, as it is seldom used in high-temperature corrosion-resistant applications. A possible exception might be short-lived instances, such as rocket components.

Fig. 3 shows common forms in which brazing alloys are supplied.

A number of other high-temperature brazing alloys are known to be in limited usage or in developmental stages. Nickel-chromium-silicon, nickel-germanium, nickel-silicon-phosphorus, and nickel-chromium-phosphorus alloys deserve mention.

Collection of significant commercial brazing applications has been attempted. Every case deals with metal components which have been assembled by high-temperature corrosion-resistant brazing methods. Where possible, a few pertinent facts will be given, including identification of the part, service requirements, and particular brazing technique employed.

It is believed that the aims of this paper will be served best by presenting only brief comments on each item as this is a broad review of high-temperature brazing applications.

Case Histories

1. *Copper Brazing* – The familiar process of copper brazing will not be emphasized because the braze metal as well as the base metal (which is usually mild or low-alloy steel) is not considered corrosion resistant. However, certain facts of this process are pertinent to our discussion.

Copper brazing is a high-temperature operation. Furnace temperatures of 2000 to 2150 F are required.

Careful control of temperature, time, and quality of the reducing atmosphere is essential.

The process has been very highly developed as a precise metallurgical production tool. Many thousands of tons of successfully copper-brazed parts have firmly established its production significance and economies.

Intricate multicomponent assemblies are copper brazed using completely automatic equipment. As a stimulating example, consider the Chevrolet torus shown in Fig. 4.

Eighty-three stamped sheet-metal parts are positioned by spot-tack welding. Brazing material is applied as a copper powder paste. Approximately 600 in. of braze fillet are generated on this particular unit.

The hydraulic torque converter history is indeed a success story for copper brazing.[10]

2. *Coast Metals* – Coast Metals, Inc., Little Ferry, N. J., has developed a series of brazing alloys. Compositions contain a minimum of 85% nickel, 2-3% boron, 3-5% silicon. One alloy, Coast

[10] See *SAE Journal*, Vol. 59, March, 1951, pp. 17-21: "Stamped Torque Converter Breeds Production Problems," by H. O. Flynn.

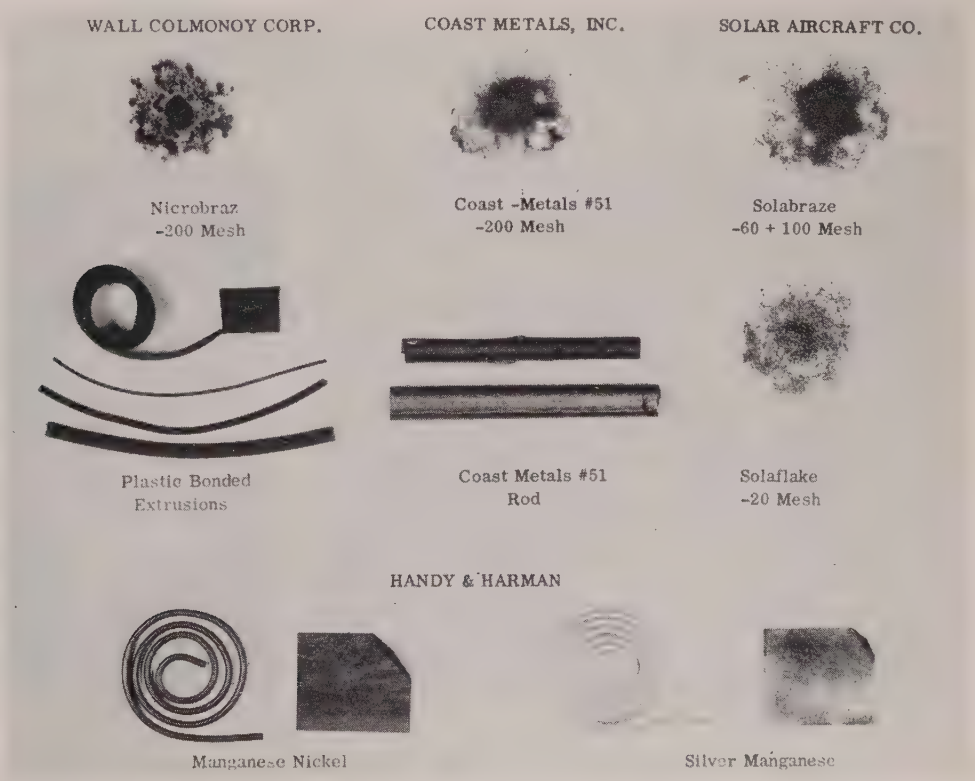
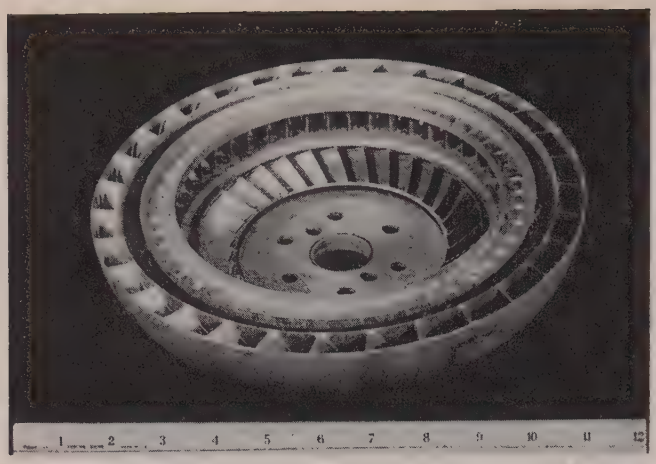
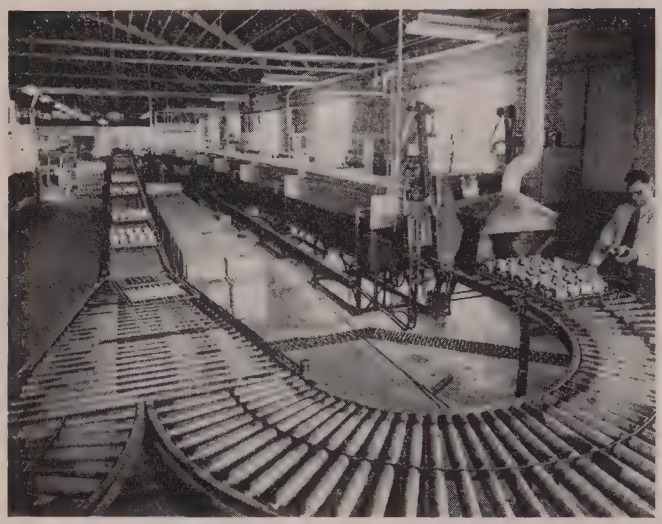


Fig. 3 – Common forms in which brazing alloys are supplied



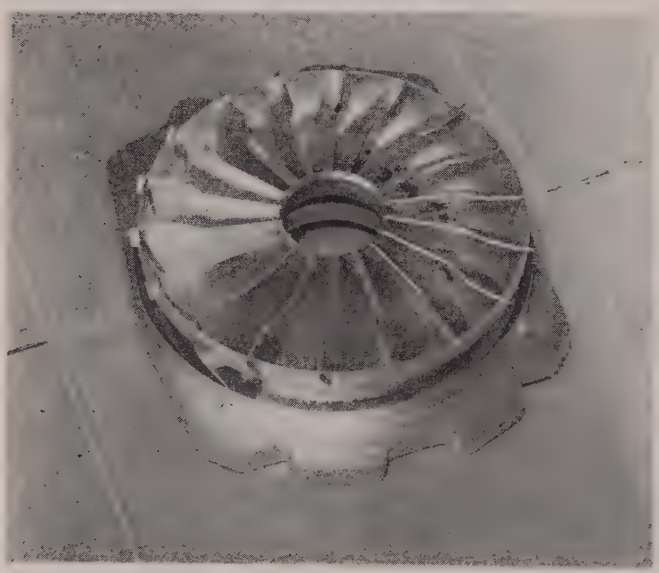
Courtesy, Fabriform Metal Products Co.

Fig. 4 – Chevrolet torus



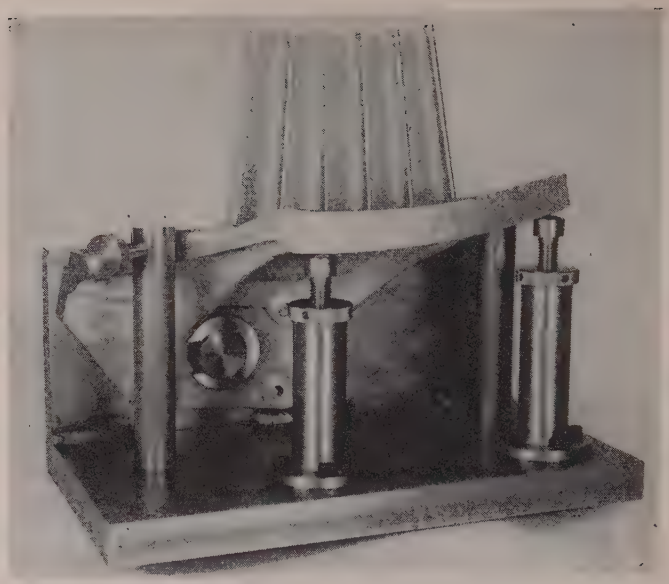
Courtesy, Superweld Corp.

Fig. 5 – Discharge end of Superweld's completely conveyorized 105-ft furnace



Courtesy, Superweld Corp.

Fig. 6 – 14-in. diameter impeller construction brazed with Coast #52

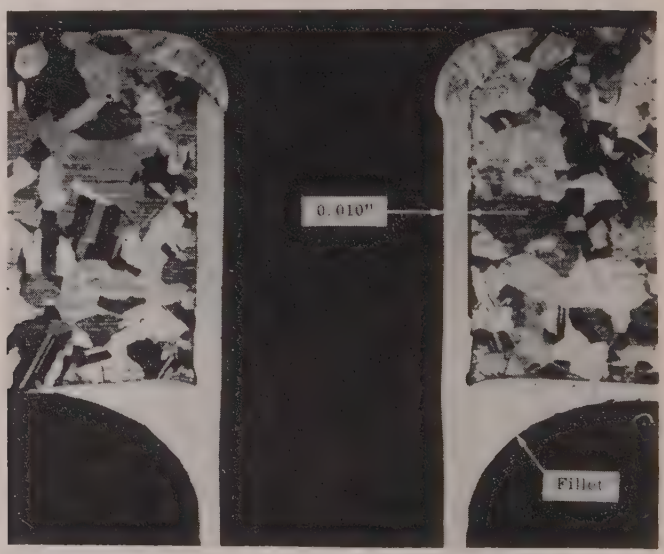


Courtesy, Wright Aeronautical Div., Curtiss-Wright Corp.

Fig. 7 – Turbine stator shroud-vane assembly jigged for silver manganese brazing in dry hydrogen

Courtesy, Rolls Royce Ltd.

Fig. 8 – "Nimonic 80" hollow gas-turbine blade



Courtesy, Metallurgy Division, Oak Ridge National Laboratory

Fig. 9 – Photomicrograph of 10-mil wall Type 316 stainless tubing cone-arc welded to ⅛-in. thick Type 304 stainless header

No. 53, contains in addition, approximately 7% chromium.

Flow point: about 1850 F.

Recommended furnace temperature: 1900-2050 F.

Braze using flux or protective atmosphere.

Available as powder or cast rod.

Outstanding development work has been performed by Superweld Corp.[11] in adapting their excellent copper-brazing facilities to accommodate the requirements of these high-temperature corrosion-resistant brazing alloys. Fig. 5 shows the discharge end of their completely conveyorized 105-ft furnace. The unit is rated at 300 kw and can process over one-half ton of parts per hour. Stainless and high-alloy steel assemblies have been brazed in this equipment using any of several Coast metals alloys.

Fig. 6 is a 14-in. diameter impeller construction brazed with Coast No. 52.

3. *Silver-Manganese* – A silver-manganese alloy is manufactured by Handy & Harman, 82 Fulton St., New York City.

85% Ag-15% Mn has the following properties:

1. Solid-solution type alloy, tough, ductile.
2. Melting point: 1750 F.
3. Flow point: 1760 F.
4. Furnace temperature required: 1850-1950 F.
5. Available as wire, strip.
6. Braze using "H" flux, or pure, dry hydrogen, –80 F dewpoint.

Application for this brazing material includes joining of stainless-steel and high-alloy parts for elevated temperature use up to about 1300 F.

Fig. 7 shows a turbine stator shroud-vane assembly jigged for silver-manganese brazing in dry hydrogen. After brazing, tolerances between blade tips and flange were well within the ±0.005 in. specified. This high degree of precision illustrates a salient feature characteristic of a well-engineered brazing application.

Other successful brazing applications using Ag-Mn include:

(a) Attachment of Inconel fuel line fittings.

(b) Joining of Stellite valve seats to stainless valve bodies.

(c) Applying of cemented carbide tool tips to steel shanks for hot machining operations.

4. *Manganese-Nickel* – A manganese-nickel alloy is manufactured by Handy & Harman, 82 Fulton St., New York City.

68% Mn-32% Ni has the following properties:

1. Solid-solution type alloy, tough.
2. Melting point: 1850 F.
3. Flow point: 1860 F.
4. Brazing conditions are similar to preceding silver-manganese alloy.
5. Alloy is subject to scaling at elevated temperatures.
6. Available as wire, powder.
7. Operating temperatures for Mn-Ni brazes range up to about 1500 F.
8. As this alloy is not attacked by molten sodium, a potential use is in liquid-cooled nuclear reactors.

5. *Palladium-Silver* – A palladium-silver alloy is manufactured by the Mond Nickel Co., Ltd., Curzon Street, London, W. I., England.

The United States representative for brazing alloys is American Platinum Works, Newark, N. J.

75% Ag-20% Pd-5% Mn has the following properties:

1. Tough, ductile alloy.
2. Brazing temperature: 2050 F.
3. Forms available: wire, strip.

Only a limited amount of information is available concerning this alloy known as "S.P.M.1."

"S.P.M.2" is another such high-temperature brazing alloy. Recommended brazing temperature is 2240 F.

Fig. 8 is a "Nimonic 80" hollow gas-turbine blade. The airfoil sheath is brazed over a large shear area to the root section.

6. *Palladium-Nickel* – An experimental palladium-nickel alloy is manufactured by the Metallurgy Division, Oak Ridge National Laboratory, Oak Ridge, Tenn.

60% Pd-40% Ni has the following properties:

1. Tough, ductile alloy, solid-solution type.
2. Flow point: about 2250 F.
3. Brazing temperature: about 2300 F.

Fig. 9 is a photomicrograph of 10-mil wall Type 316 stainless steel tubing attached to a ⅛ in. thick Type 304 stainless steel header by "cone-arc" welding.[12] In order to improve the strength properties of such welded assemblies, palladium-nickel brazing alloy was subsequently flowed around the opposite face. Substantial diffusion of the braze with the base metal is evident.

7. *Solabraze* – Solabraze is manufactured by Solar Aircraft Co., 2200 Pacific Highway, San Diego, Calif.

A number of alloys are currently being developed to meet specific classes of high-temperature brazing applications. Compositional ranges are in general 60 to 85% nickel, up to 20% chromium, up to 30% manganese, ½ to 5% silicon, 0 to 3% boron, together with varying amounts of iron and molybdenum.

Flow point: from 1750 to 2000 F.

Forms now being used: powder and cast rod.

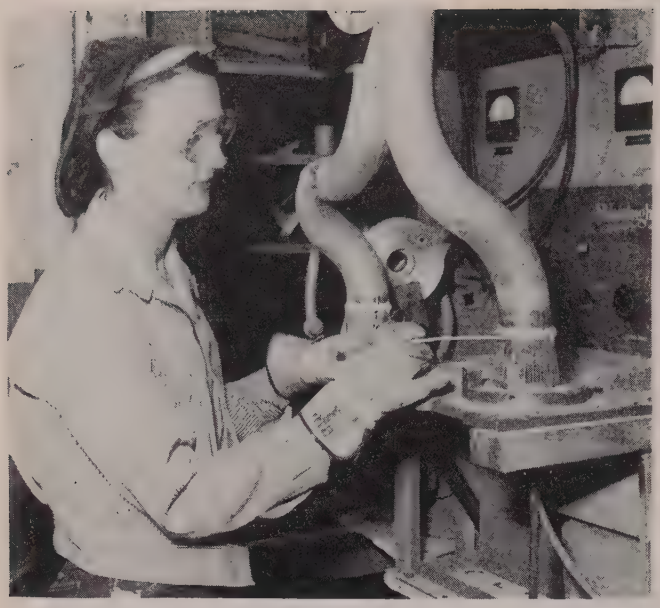
Braze using flux or pure, dry hydrogen.

For many years Solar has been aggressively developing and manufacturing exhaust manifolds for aircraft reciprocating engines. Suitable high-temperature corrosion-resistant brazing methods for attaching manifold flanges have been placed in production. The family of brazing alloys used, termed "Solabraze," was evolved as solutions to the problems of brazing such joints.

Fig. 10 is a representative picture of an actual

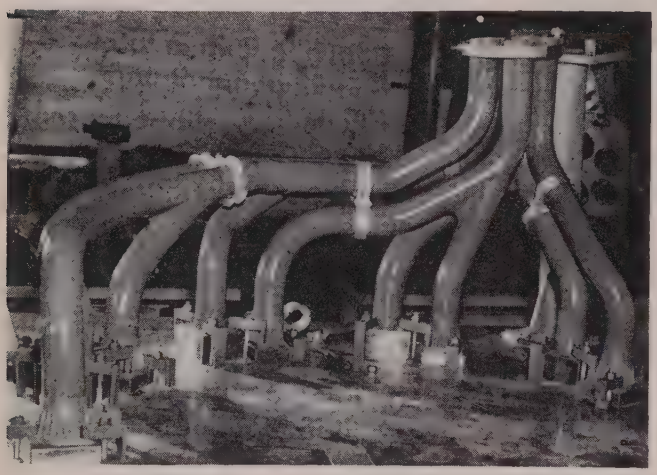
[11] Superweld Corp., 6840 Vineland Ave., North Hollywood, Calif.

[12] See *The Welding Journal*, Vol. 32, No. 7, July, 1953, pp. 597-602: "Cone Arc Welding," by P. Patriarca and G. M. Slaughter.



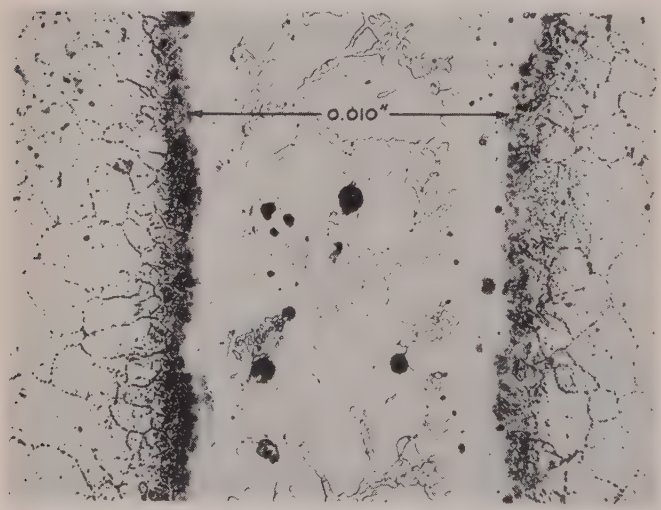
Courtesy, Solar Aircraft Co.

Fig. 10 – Actual brazing operation



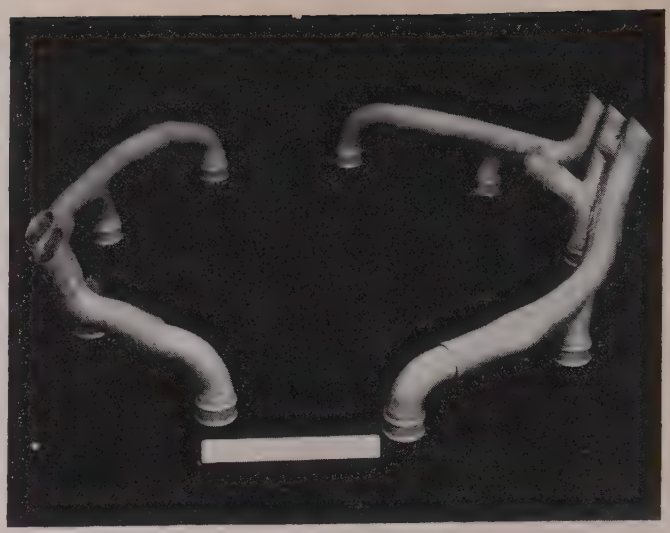
Courtesy, Solar Aircraft Co.

Fig. 11 – Half manifold set arranged on inspection jig



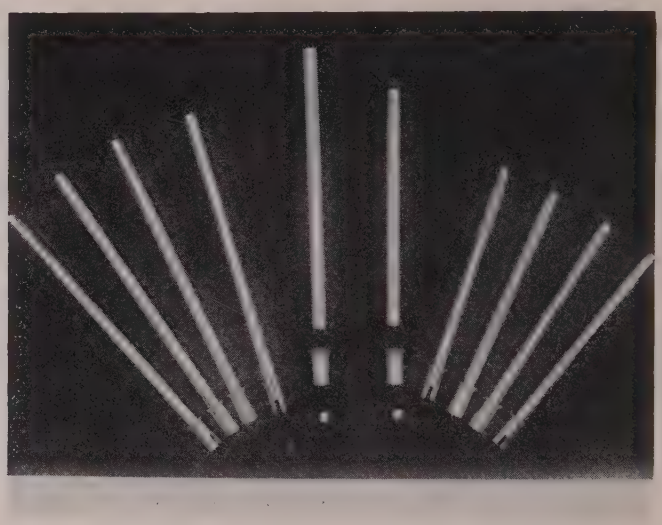
Courtesy, Solar Aircraft Co.

Fig. 12 – Photomicrograph of sectioned and etched sheet metal flange joint after 500-hr dynamic cycling to 1500 F



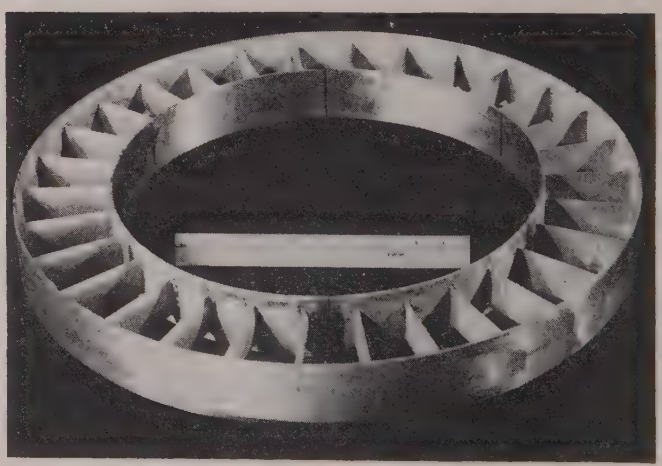
Courtesy, Solar Aircraft Co.

Fig. 13 – Flange fittings attached by brazing to cylinder port legs of this exhaust manifold



Courtesy, Solar Aircraft Co.

Fig. 14 – Afterburner fuel spray nozzles



Courtesy, Solar Aircraft Co.

Fig. 15 – 24-in. OD vane-shroud Solabrazed assembly

brazing operation. A stable refractory flux is necessary to permit braze metal flow in air at 2000 F. Automatic time-temperature programming control is used for the 2½-min. heating cycle.

Fig. 11 shows a half manifold set arranged on an inspection jig. Both AISI 347 cast flanges as well as AISI 321 sheet metal flanges are used in this brazed assembly. After brazing, the manifolds are ceramic coated inside and out by the Solaramic[13] process. This accounts for the high reflectivity observed in the photograph.

Fig. 12 is a photomicrograph of a sectioned and etched sheet metal flange joint after 500-hr dynamic cycling to 1500 F. The massive braze metal measures approximately 0.010 in. in radial thickness. Diffusion with the base metal is indicated by the dark parallel bands within the 321 base metal. A dense braze structure is evident consisting of two well-defined phases. A few scattered voids are typical of massive brazes made using a flux rather than, for example, a pure, dry hydrogen atmosphere.

Fig. 13 shows another exhaust manifold. The nine cylinder port legs in this instance have identical flange fittings attached by brazing. Operating temperatures at the joint are in the range of 1000 to 1300 F.

Afterburner fuel spray nozzles are presented in Fig. 14. These Type 321 stainless steel assemblies inject additional fuel directly into the jet-engine exhaust stream. The two exploded views show the orifice and sleeve which are high-temperature brazed to the heavy-walled tube in one operation.

An outstanding field for application of high-temperature corrosion-resistant brazing is turning and straightening vane assemblies for use in turbojet engines.

Fig. 15 shows a 24 in. OD vane-shroud Solabrazed assembly currently under development.

A useful characteristic of the brazing process is of great significance in such large multicomponent parts. Overall distortion during brazing can be maintained within remarkably small limits. It is not unreasonable, for example, to maintain flatness over a 2-ft diameter within 0.030 in.

8. *Nicrobraz* – Nicrobraz is manufactured by Wall Colmonoy Corp., John R. St., Detroit 3, Mich.

Hard, highly corrosion-resistant alloy.

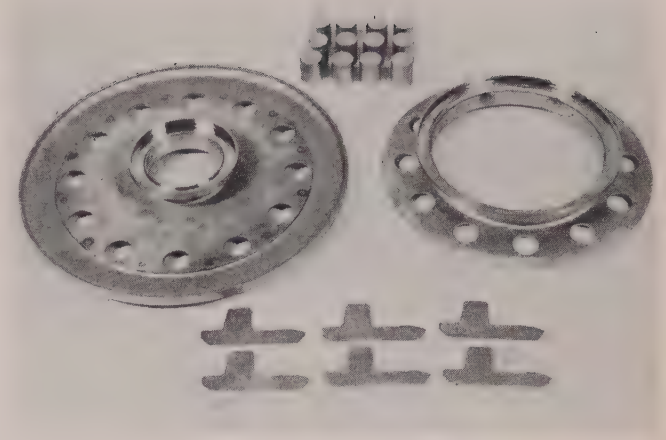
Flow point: about 1930 F.

Brazing temperature: 1950-2150 F.

"Nicrobraz" is produced in accordance with Aeronautical Material Specification 4775.[14] Nominal alloying composition of this nickel-base brazing alloy is: 4% silicon, 16.5% chromium, 4% iron, 3.8% boron, and 1% carbon.

[13] "Solaramic," trademark for ceramic coatings and processes developed by Solar Aircraft Co., San Diego, Calif.

[14] See SAE Aeronautical Material Specifications AMS 2675, "Nickel Alloy Brazing" and AMS 4775, "Brazing Alloy, Nickel Base—4 SI, 16.5 CR, 4 FE, 3.8 B, 1.C"



Courtesy, Wright Aeronautical Div., Curtiss-Wright Corp.

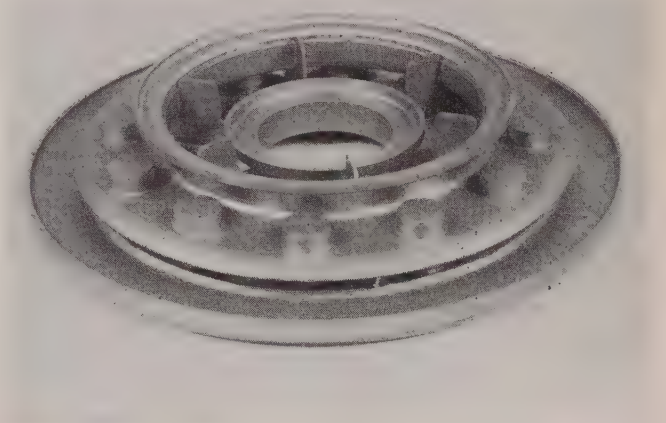
Fig. 16 – Stainless steel parts for a heat shield

From the earliest applied development work in the field of corrosion-resistant high-temperature brazing to the present, Nicrobraz has been used extensively.

It was probably the first commercially available brazing alloy to exploit another rather unique characteristic. Nicrobraz joints can be made which exhibit useful strengths at temperatures in excess of the melting point of the massive brazing alloy itself. This significant fact has an interesting explanation.

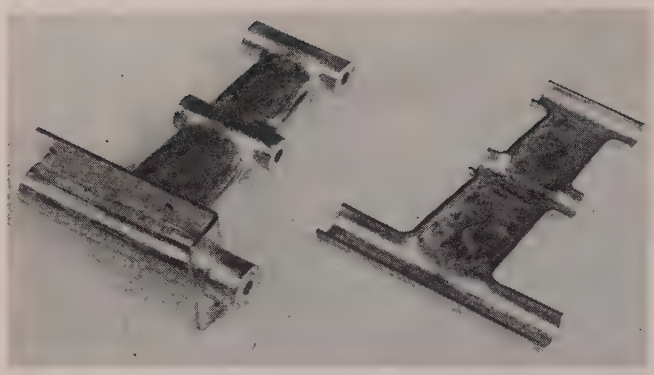
With relatively tight fitting, brazed joints, the effects due to atomic diffusion at the brazing temperature occur very rapidly. For example, in the case of stainless steel joined with Nicrobraz, mutual diffusion between the base metal and braze metal produces an entirely new series of alloys in and adjacent to the joint. A Nicrobraz composition with boron, carbon, and silicon removed would melt at about 2550 F. Diffusion during brazing or by subsequent heat-treating simply reduces the concentration of these critical elements and so raises the "remelt" temperature of the joint.

In addition to this characteristic of Nicrobraz,



Courtesy, Wright Aeronautical Div., Curtiss-Wright Corp.

Fig. 17 – Completed heat shield from parts shown in Fig. 16

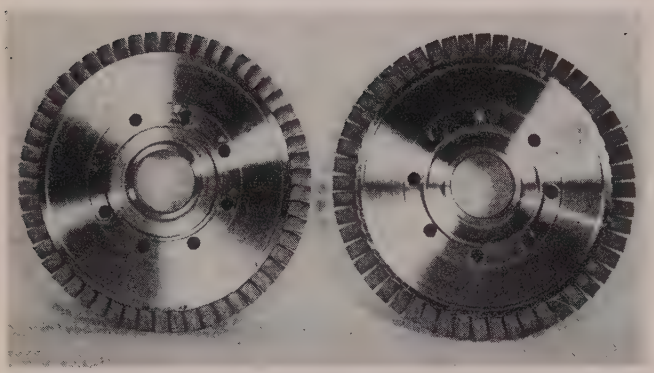


Courtesy, Wall Colmonoy Corp.

Fig. 18 – Fuel-injection-nozzle support

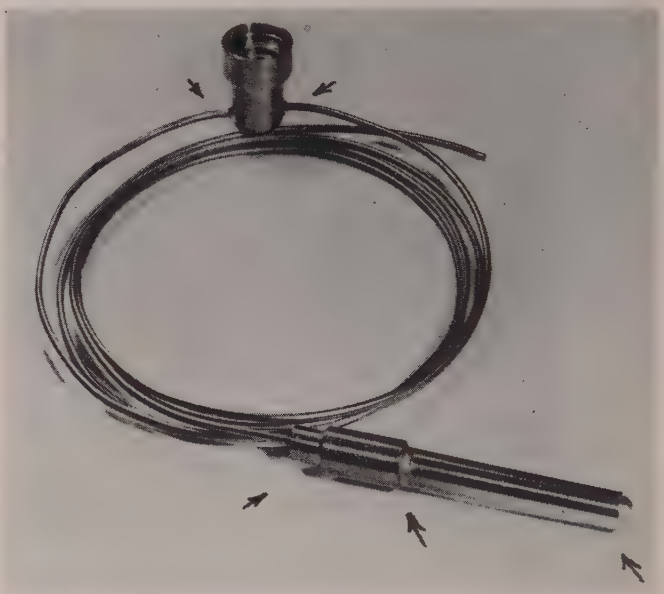
Courtesy, Wall Colmonoy Corp.

Fig. 19 – Nicrobrazed turbine nozzle diaphragm or stator section



Courtesy, Wright Aeronautical Div., Curtiss-Wright Corp.

Fig. 20 – Blades of this compressor rotor wheel were Nicrobrazed to disc periphery



Courtesy, Wall Colmonoy Corp.

Fig. 21 – Five Nicrobrazed joints are made in one furnace operation for this mercury control element

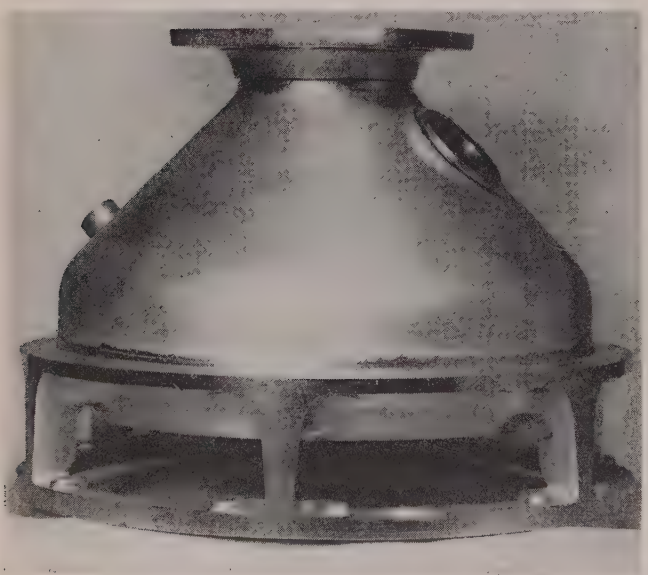
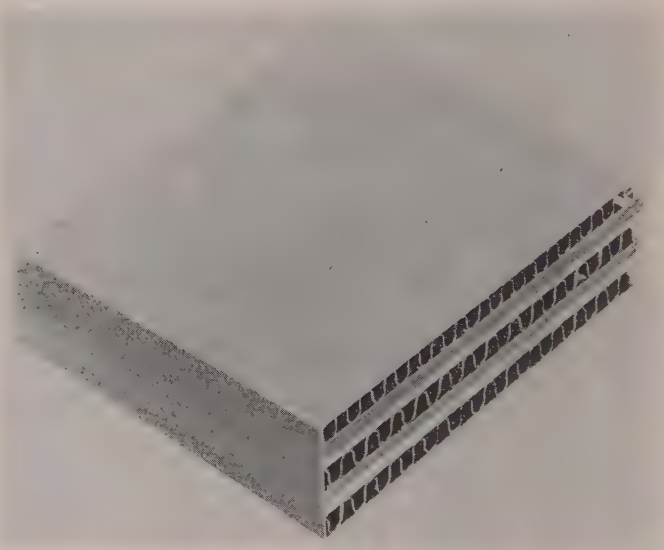
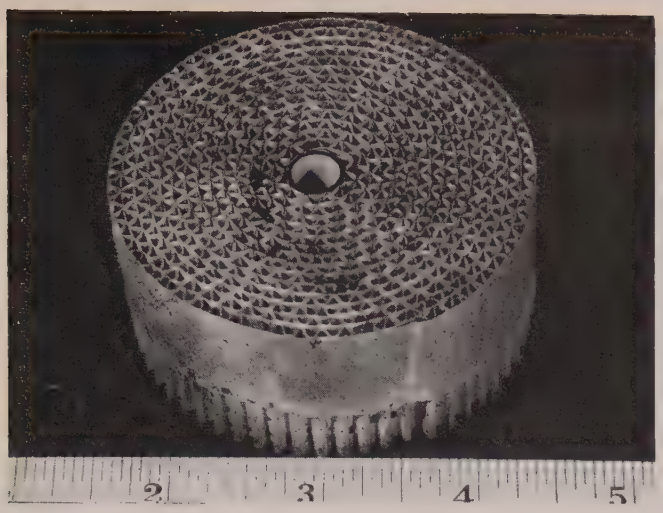


Fig. 22 – Combustion chamber serves as turbojet starting mechanism (Nicrobrazed by Wall Colmonoy Corp. for Bendix Aviation Corp.)



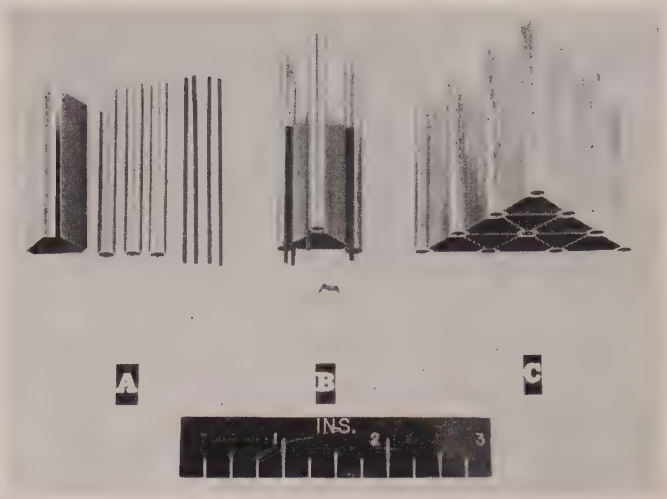
Courtesy, Wall Colmonoy Corp.

Fig. 23 – Cross-flow, corrugated-type heat exchanger



Courtesy, Solar Aircraft Co.

Fig. 24 – Experimental rotary regenerator



Courtesy, Metallurgy Division, Oak Ridge National Laboratory

Fig. 25 – Development of heat exchanger requiring precision brazing of complex parts

its outstanding oxidation and corrosion resistance properties are important.

Nicrobraz is being used by a number of manufacturers to produce brazed assemblies, and a few of the many known applications are presented.

Fig. 16 represents 23 stainless steel parts which are assembled to form a heat shield. Fixturing is minimized by the use of spot and tack welds. The completed unit is shown in Fig. 17. All joints are brazed with Nicrobraz in one furnace operation. Satisfactory service operation at 1000 F is reported.

Fig. 18 is a fuel-injection-nozzle support. A S-590 alloy half stamping is shown, together with the brazed assembly comprising stampings, plugs, and top shelf.

Fig. 19 presents a Nicrobrazed turbine nozzle diaphragm or stator section. Shroud is Type 321 stainless steel, vanes are cobalt base alloy. These large precision assemblies have been produced in quantity, evidence of a highly developed technique.

Another outstanding application of Nicrobraz is presented in Fig. 20. This is a 12-in. diameter AISI 431 compressor rotor wheel designed to operate at 12,000 rpm and 500 F maximum. Blades were attached to the disc periphery in a dry hydrogen furnace brazing process.

A rather unusual application, shown in Fig. 21, is a mercury control element in Type 347 stainless steel. Five Nicrobrazed joints are made in one furnace operation.

As Nicrobraz is resistant to mercury attack at temperatures up to 1700 F, it offers interesting possibilities in the high-temperature instrumentation field.

A beautifully engineered assembly is the combustion chamber pictured in Fig. 22. This heavy-duty unit functions as a turbojet starting mechanism. Construction details are not available.

Commercial production is reported.

Special Structures

Previous discussion described various commercial high-temperature brazing alloys and representative applications for each alloy.

This section will be confined to a presentation of several laboratory type developments that may presage new fields of applications for high-temperature materials. Emphasis is given to two categories: (1) heat exchangers, and (2) high strength-to-weight ratio structures.

Both classes require the utilization of very thin sheet metal components assembled into a three-dimensional, stiff structure. Whereas the efficient high-temperature heat exchanger must transmit thermal energy at a prodigious rate, anticipated uses for high strength/weight structures favor thermal barrier or insulating type designs.

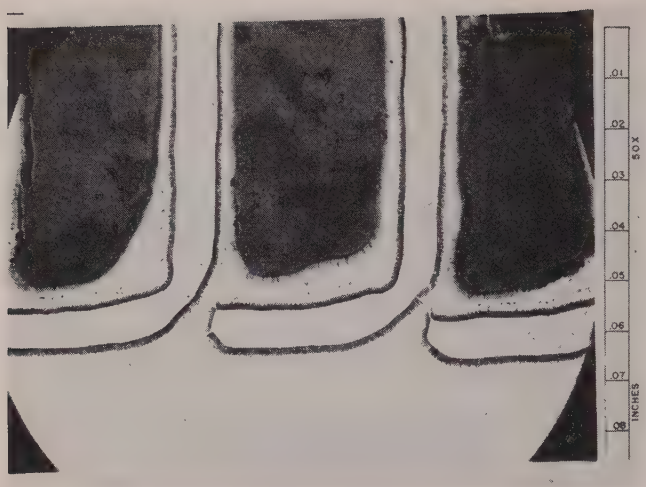
Heat Exchangers – In general, heat exchangers are required for handling of fused salts, molten metals, and hot gases, in conjunction with liquid metals, air, or water as coolant. Industrial uses involve chemical and metallurgical processing operations. Transfer of heat as a means of utilization of atomic energy for power development is also of vital concern.

Fig. 23 shows a cross-flow, corrugated-type heat exchanger. Material is 0.010-in. AISI 316 stainless steel brazed in one operation with Nicrobraz.

Heat exchangers measuring up to 8 in. square by 15 in. high have been fabricated by Wall Colmonoy Corp.

Fig. 24 represents an experimental rotary regenerator. Material used was AISI 321 stainless steel.

An unusual heat-exchanger development requiring precision brazing of complex components is



Courtesy, Metallurgy Division, Oak Ridge National Laboratory

Fig. 26 – Photomicrograph of radial fin-tube type of heat exchanger

presented in Fig. 25. Progressive steps leading to the delta unit assembly are indicated by A, B, and C.

Materials of construction consist of 0.010-in. copper clad on both sides by 0.002-in. Inconel. The Inconel tubes are 3/16 in. OD by 0.020 in. wall.

A modified Nicrobraz alloy was used for furnace brazing at 1920 F, only 60 deg below the melting point of copper.

A more conventional radial fin-tube type of heat-exchanger construction is indicated by the photomicrograph in Fig. 26. Punched discs of Inconel-clad copper are stacked about the 3/16-in. Inconel tubes. Spacing is neatly accomplished by the protruding lip formed during a stamping operation.

Again, brazing is accomplished using the modified Nicrobraz alloy under pure, dry hydrogen brazing conditions.

Design operating temperature for these brazed heat-exchanger assemblies is in the 1300 to 1650 F range.

High-temperature brazing is also being successfully employed to assemble hollow turbine guide vanes. Fig. 27 and 28 represent sections from several sheet metal vanes that are designed for internal fluid cooling. The function of the corrugated foils in Fig. 27 is nonstructural and is simply to provide additional heat-transfer area.

In Fig. 28 the core also functions as a stiffener. For selective dissipation of heat at both leading and trailing edges, massive cast-copper fillets can be employed as shown.

A number of governmental as well as private agencies are attempting to develop similar types of brazed constructions for highly stressed parts, particularly turbine rotor blading. Referring back to Fig. 8, one possible solution to this particular problem is indicated.

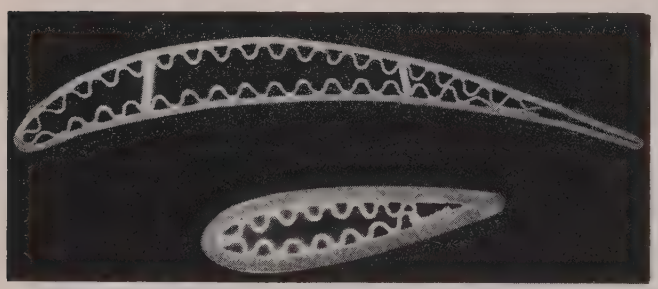
High Strength-to-Weight Ratio Structures – Great interest is currently being shown in a new material of construction for high-temperature use. This is an all-metal sandwich material of honeycomb or corrugated core designed to fill the need for stiff, high strength-to-weight ratio structures that maintain strength at elevated temperatures.

High-speed aircraft and missile applications under study include firewalls, tailpipe shroud covers, engine compartments, hot gas ducting, control surfaces, leading edges, and nose sections, to mention but a few.

Wright Air Development Center has initiated several investigations and production projects to stimulate developments in this field. Persons familiar with the numerous uses of resin-bonded aluminum honeycomb structures will immediately appreciate a similar structure but of higher strength and capable of sustained operation at say 600 to 1200 F.[15]

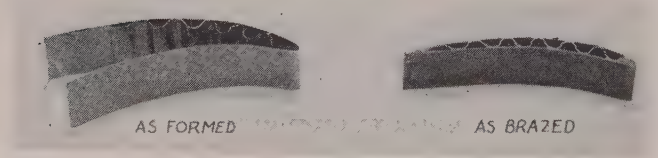
From a materials standpoint, the relationship of physical properties per unit weight for the temperature range desired is fundamental. Fig. 29 presents such data graphically for a variety of different metals available in sheet form. These curves are arbitrarily based on short-time 0.2%-offset yield strengths. The so-called light-weight alloys are shown dashed for easier identification.

A study of this figure emphasizes the importance of utilizing precipitation hardening alloys of the "Armco 17-7 PH" type for maximum oper-



Courtesy, Solar Aircraft Co.

Fig. 27 – Sheet metal vanes designed for internal fluid cooling. Nonstructural corrugated foils provide additional heat-transfer area



Courtesy, Solar Aircraft Co.

Fig. 28 – Sheet metal vane where core also functions as a stiffener

[15] See WADC Unclassified Report, Dec. 15-16, 1952, "Papers Presented at Air Force-Aircraft Industries Association Conference on Structural Adhesives for Metals and Sandwich Construction," Wright Air Development Center, Dayton, Ohio.

ating temperature below about 800 F. At higher temperature, "Inconel X" is outstanding.

Assemblies of these high-temperature, high-strength alloys can be securely joined when high-temperature corrosion-resistant brazing methods are employed.

Fig. 30 represents honeycomb core stock. Metal gage is 0.002 in., and cell nodes are joined by small spot welds in "Squarcel,"[16] left, while "Hexcel,"[17] at the right of the photograph, has plastic bonded nodes.

The fan-like section represents the gradual expanding of honeycomb from the completely compressed state.

Several high-temperature brazed structures are assembled in Fig. 31. Airfoil, curved panel, and duct section indicate the versatility to be expected from such sandwich material. Both core and face material is 0.002-in. AISI 321 stainless-steel sheet. Application is primarily as very-light-weight, high-temperature, self-supporting heat shrouds or insulation.

Because of the extremely thin faces, slight surface deformation produces a tracing of the core beneath. A portion of core has been sawed and bent back exposing the internal character for inspection.

Approximately 4000 in. of braze-metal fillet is required for 1 sq ft of this particular honeycomb sample! Total weights of 0.3-0.46 psf have been

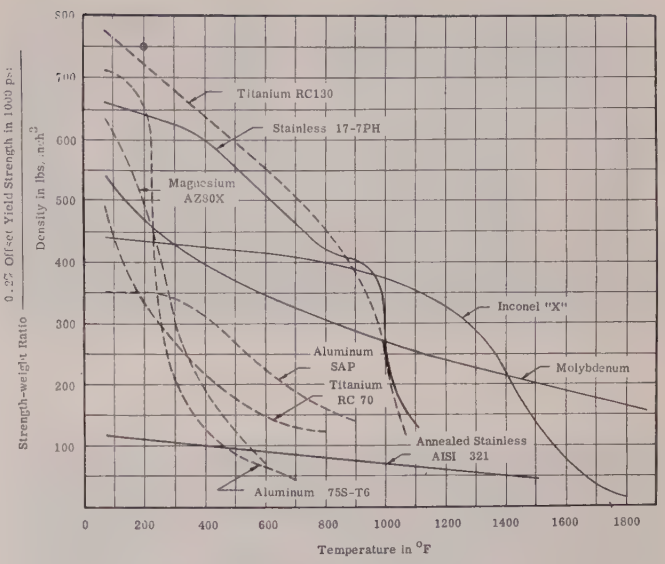


Fig. 29 – Strength-weight ratio versus temperature

measured for this particular type of all-metal sandwich. It is of interest to note that braze metal may account for 25% of the total weight.

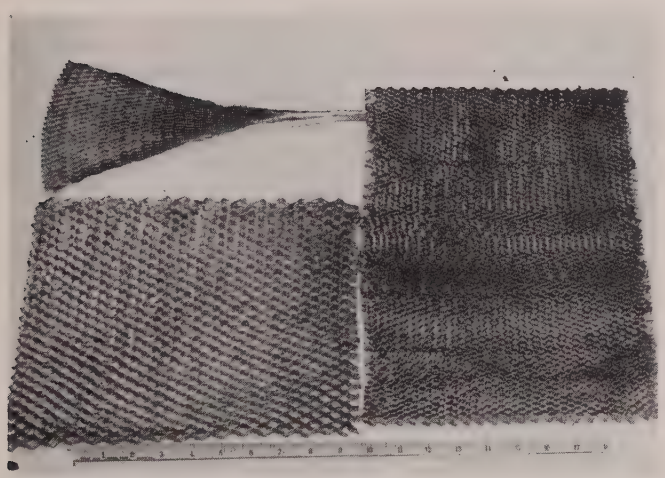
For improved corrosion resistance, a thin Solaramic ceramic coating has been applied to the tubular duct sample.[18] If high-strength, low-alloy steels such as SAE 4130 are used in sandwich construction, such protective measures may be mandatory.

Fig. 32 shows a typical thickness cross-section. Note the small, uniform fillets at the bases of each column. Two thicknesses of face sheeting make a

[16] "Squarcel," trademark of John J. Foster Co., 19666 South Harbor Blvd., Santa Ana, Calif.

[17] "Hexel," trademark of California Reinforced Plastics Co., 951 61st St., Oakland 8, Calif.

[18] See "Ceramic Coatings Solve High-Temperature Problems," by John V. Long. Paper presented at SAE Aeronautic Meeting, Los Angeles, Oct. 3, 1952.

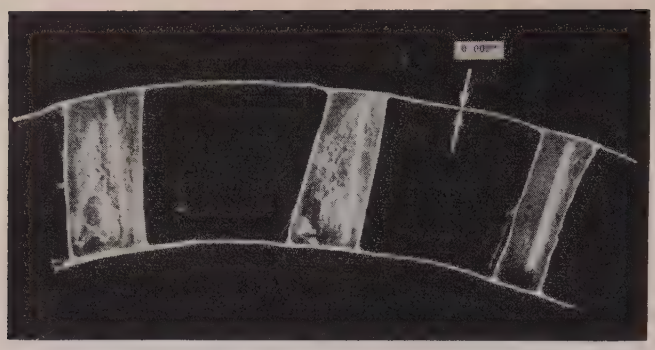


Courtesy, John J. Foster Co.

Fig. 30 – Honeycomb core stock "Squarcel" (left) and "Hexcel" (right)

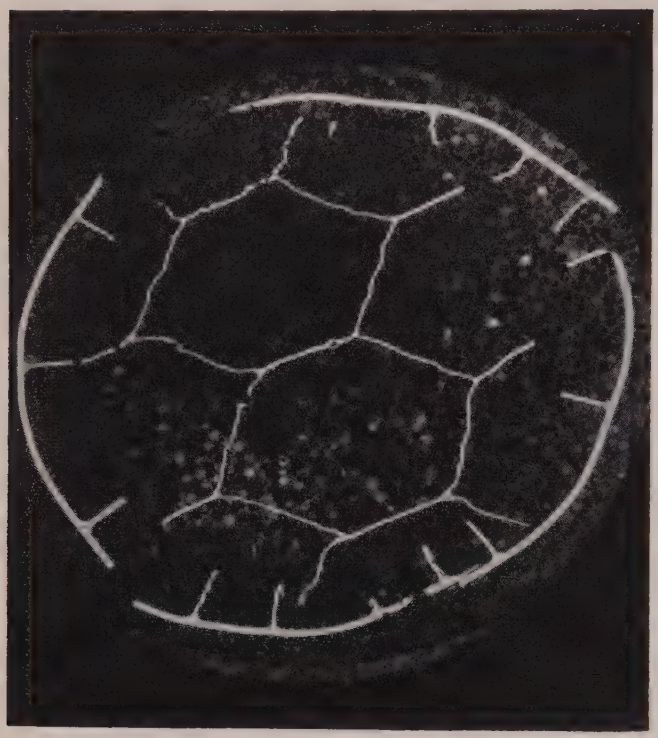
Courtesy, Solar Aircraft Co.

Fig. 31 – High-temperature brazed structures: airfoil, curved panel, duct section



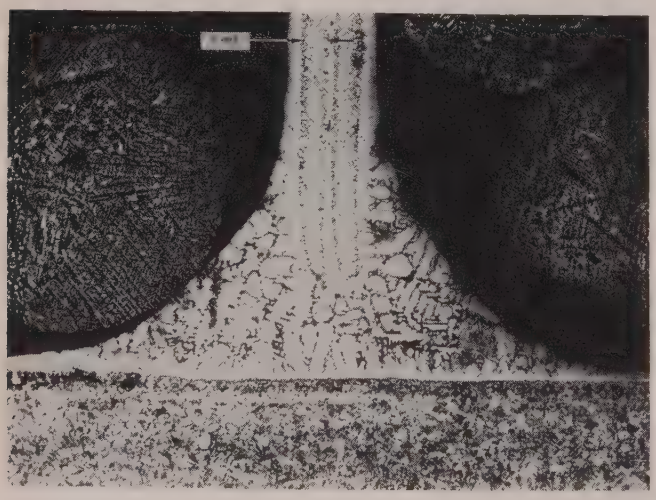
Courtesy, Solar Aircraft Co.

Fig. 32 – Typical thickness cross-section



Courtesy, Solar Aircraft Co.

Fig. 33 – Polished and etched foil cross-section embedded in mixture of steel shot and phenolic mounting resin



Courtesy, Solar Aircraft Co.

Fig. 34 – Representative joint between honeycomb core and face sheet

nice lap joint as may be noted in the lower left-hand corner.

In order to metallographically examine cross-sections of such thin foil structures, a uniform edge supporting means is necessary. Excellent results have been obtained by embedding the desired section in a mixture of steel shot and phenolic mounting resin. Such a sample after polishing and etching is presented in Fig. 33.

A bridged joint between honeycomb core and face sheet is shown in Fig. 34. In this photograph the face material is 0.010 in. and two 0.002-in. core nodes are seen. Braze flow is observed to be complete with excellent bonding of abutting nodes. This latter characteristic is particularly essential if resin-bonded honeycomb core stock is used, as in Fig. 33. During brazing, the resin decomposes and must be replaced by braze metal if maximum physical properties are to be realized.

Light-weight, all-metal honeycomb sandwich must have a minimum of braze metal and yet no unbonded areas. Maintenance of metal-to-metal contact between faces and core during brazing is a basic problem. This requires very close tolerances on core thickness and furnace jigging, as well as uniform pressure distribution throughout the braze cycle. New fabrication techniques will certainly require "imagineering"!

Conclusions

An attempt has been made to present an objective review of uses for high-temperature corrosion-resistant brazing.

Many structural parts which we have briefly discussed demonstrate the advantages and potentialities of the brazing process.

Complex and multicomponent sheet-metal parts lend themselves to assembly by this method.

High-strength joints can be made between cast and sheet metal, dissimilar metals, widely dissimilar metal thicknesses, and other unconventional combinations.

High-temperature, corrosion-resistant brazing is rapidly becoming a major manufacturing operation. Intelligent application of this new technology is successfully meeting stringent present-day demands. Future automotive, aircraft, and missile designs will rely heavily on this efficient process of metal joining.

Acknowledgment

The authors wish to express their thanks to the persons and organizations whose contributions appear in this paper.

Appreciation is proffered also to those who submitted pertinent information of a general nature.

Solar Aircraft Co. furnished the illustrations and data concerning "Solabraze." We are grateful to them for the unique opportunity to present this information as part of our overall review of the high-temperature brazing field.

A New Concept of Explosion Testing of Aircraft Equipment

W. K. Klose, Boeing Airplane Co.

This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 2, 1953.

IN explosion testing to determine the safety of equipment for airplane applications, certain aspects of explosion phenomena are often overlooked.

We first became aware of this some months ago when a fuel boost pump, under test for compliance with the Air Force Procedure II Explosion Proof Test, unexpectedly ignited the external explosive mixture: this, after successfully containing 140 internal test explosions. Every subsequent effort using the specification test methods failed to reproduce the explosion.

Confronted with such a situation, what should the test engineers conclude? Was the pump safe or was it unsafe? Production schedules made it desirable that the pump be approved, but how could the pump be approved when it appeared impossible to duplicate the one explosion? That one explosion still remained as evidence prohibiting approval.

What follows is a summary of the findings from the research investigation which resulted from this occurrence.

A NEW procedure for explosion testing of aircraft equipment is such that the effects of mixture ratio, ignition location, and temperature are now reflected in the test results.

The investigation of the effects of these items on flame suppression, which prompted adoption of the new test procedure, is described in this paper.

The Author

W. K. KLOSE is supervisor of Powerplant Research and Development for Boeing Airplane Co. He studied engineering at the University of Nebraska, and for 12 years was engaged in manufacturing clay products at Missoula, Montana. Mr. Klose joined Boeing in 1941.

Research Program

The first part of the investigation constituted a review of available literature on flame travel through restricted openings. This, at the time, failed to reveal anything except that the pump passages as designed should successfully quench the flame.

Since this line of investigation proved of no avail it was decided to make a study of basic explosion phenomena. Literature on this subject helped in this way – it was learned at least why few people understand what to expect from the ignition of an explosive mixture: the factors involved and the chain of events as the explosion progresses become very complex. From the investigation, however, it was decided that certain factors should become very apparent in carefully controlled explosion tests.

Most important of these were the following:

1. In combustion of explosive gases in closed chambers, the temperatures attained by the burned

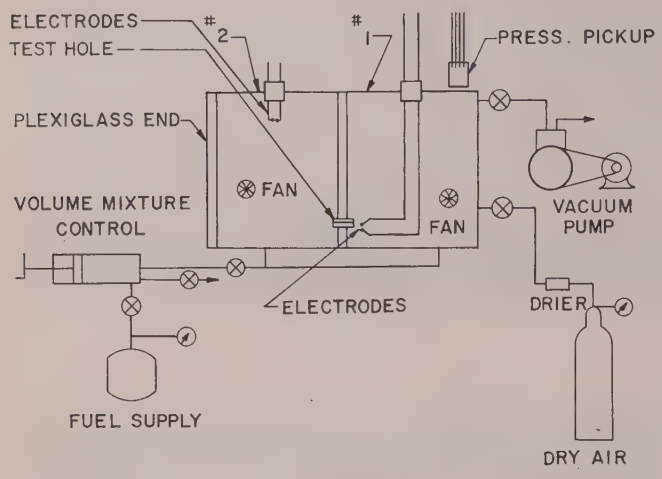


Fig. 1 – Schematic diagram of small hole flame-suppression test setup

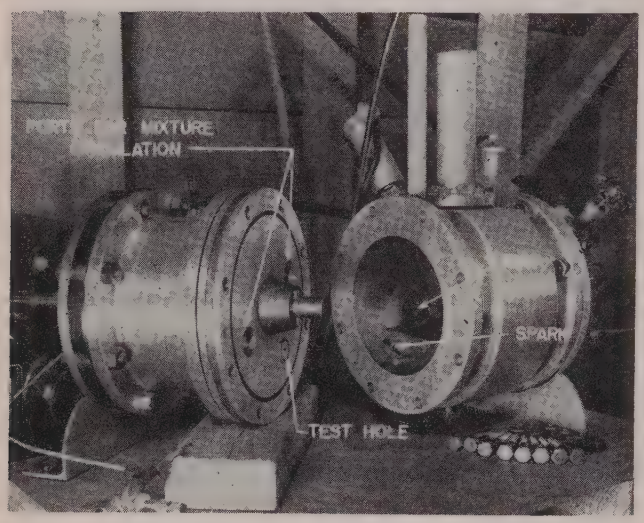


Fig. 2 – Interconnected explosion bombs

Table 1 – Effect of Ignition Location and Fuel/Air Ratio on Flame Suppression Through a Small Hole

	Configuration 1[a]		Configuration 2[b]		Configuration 3[c]	
Mixture	Tests Made	Number Failed	Tests Made	Number Failed	Tests Made	Number Failed
1.75	0	0	5	0	0	0
2.00	3	2	3	0	2	0
2.25	1	1	2	0	0	0
2.50	4	2	3	0	1	0
2.75	2	2	1	0	2	0
3.00	5	3	0	0	7	3
3.25	1	1	1	0	2	2
3.50	5	4	0	0	0	0
3.75	0	0	1	0	2	1
4.00	2	0	2	0	1	0
4.25	0	0	0	0	0	0
4.50	2	0	1	0	0	0
4.75	0	0	0	0	0	0
5.00	0	0	1	0	0	0
Total	25	15	20	0	17	6

[a] ½ in. from hole.
[b] 5 in. from hole.
[c] ½ in. from hole, displaced ½ in. to side.

gases in the immediate vicinity of the ignition source exceed those in the other regions by many hundreds of degrees (in some cases as much as 900 F).

2. The fuel/air ratio affects the flame speed and should, therefore, also affect the flame travel through suppressing devices.

3. The initial temperatures of the explosive gases affect the ease of ignition.

4. The hydrogen/carbon ratio of the gases affects the flame speed of the burning mixture and should affect the flame travel through passages.

5. Theory indicates that heat transfer to the wall affects flame travel through the passage. Condensed water on these surfaces should then affect the flame travel.

The second phase of the investigation therefore was a test approach intended to reveal the magnitude of the five effects shown.

A schematic diagram of the test setup is shown in Fig. 1, and a photograph of the explosion bombs used is shown in Fig. 2. The arrangement of the two interconnected bombs permitted insertion of the flame-suppressing device between two explosive mixtures.

The flame-suppressing devices used in this series of tests were metal plugs of various lengths with a single drilled flame hole through the center of each plug. Table 1, illustrates the effect of both ignition location and fuel/air ratio in the case of a hole whose length and diameter relationship (0.094 in. diameter and 0.79 in. length) approximated one of the vent holes in the fuel boost pump. The table shows that with the ignition source directly in front of the holes (approximately ½ in. from hole), out of 25 test explosions, 15, or 55%, exploded the external mixture. Of the 17 tries between a fuel/air mixture ratio of 2.5 to 4% (by volume), 12, or 71%, of the explosions carried through the hole.

With the ignition source 5 in. from the hole, out of 20 test explosions, no external explosions occurred.

Shown also are the results of having the ignition again placed ½ in. from the hole but displaced to one side approximately ½ in. Out of 17 test explosions, 6, or 35% fired through.

Figs. 3 and 4 are Schlieren frames taken from a high-speed movie film showing effect of ignition location on flame suppression through a small hole. In the pictures on the left, flame propagated through the hole. In the right-hand pictures, flame failed to propagate because of a change in ignition location.

Fig. 5 shows a typical curve for the flame speed for hydrocarbon-air mixtures. It shows the maximum flame speed as occurring in the region slightly above stoiciometric. This is also the region in which Table 1 shows the greatest percentage of explosions carried through the hole.

That the initial temperature of the gases affect the flame propagation can be seen from Fig. 6. Here are shown the limiting hole diameters versus length. The upper line was established as a limit-

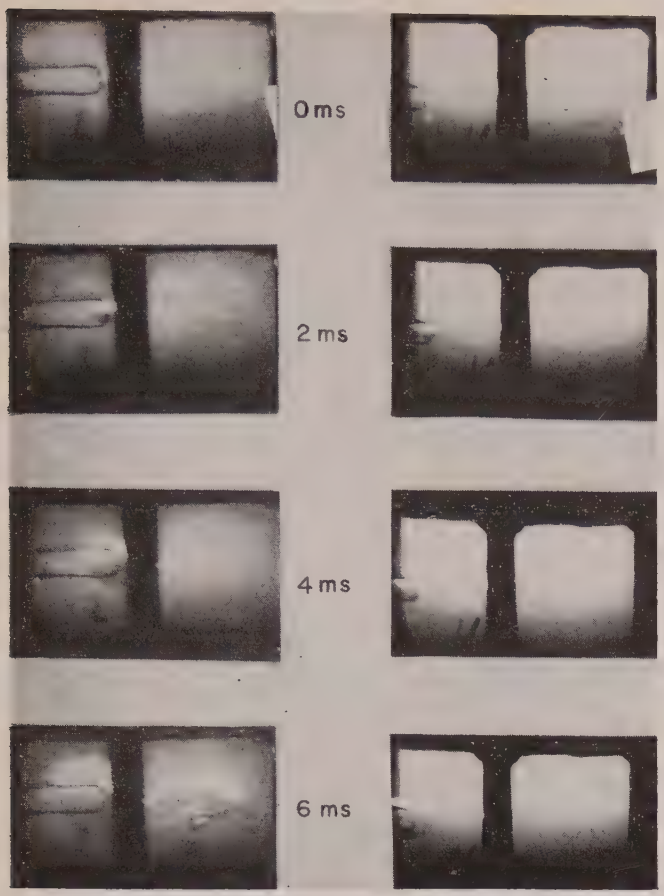


Fig. 3 – Effect of ignition location on flame suppression through a small hole

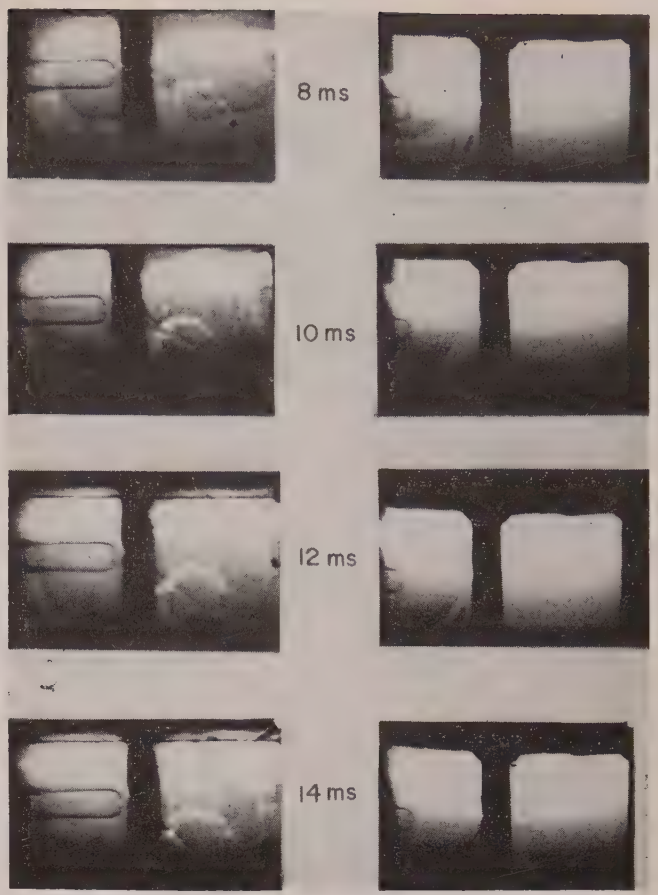


Fig. 4 – Effect of ignition location on flame suppression through a small hole

ing line for holes safe at room temperature, and the lower line establishes the holes which were safe at temperatures of 200 F.

The hydrogen/carbon-ratio effect mentioned as item 4 did not become too apparent in the tests made since no extreme ranges of fuels were tested. It was noted, however, that test results were more consistent with the use of butane as a fuel than when normal hexane or aviation gasoline were used.

That the existence of water vapor affected flame travel through suppressing devices was noted in two ways. In the case of a marginal hole (one that would suppress a flame approximately 90% of the time at room temperature and 50% humidity), this same hole, if wiped dry just prior to the test, would fire consistently.

The second case was a hole of a size and length such that 90% of the explosions propagated. This hole could be made 100% safe by running a damp pipe cleaner through the hole just prior to the bomb ignition.

Since these tests were performed on a test device in which the instantaneous pressure drop across the flame suppressing device was of the order of 60 psi, the question can be asked, "Are the results representative of what can be expected from an internal explosion in aircraft equipment?"

It was at this point in the investigation that the fuel boost pump was again checked for explosion hazard. Certain changes were, however, made in the test configuration. These changes consisted of numerous spark locations as shown in the schematic diagram in Fig. 7. A heat lamp was also added to the explosion chamber to permit the test to be performed at elevated temperatures. Fig. 8 shows the test setup.

The tests were performed at approximately 100 F, and the results are tabulated in Table 2. Out of the 58 explosion attempts, five ignited the external mixture. One configuration with the spark at 1a and 4 fired through twice out of eight tries or 25% of the time.

The vent-hole sizes as used in the original pump had been found to be 100% unsafe in the laboratory test bombs. Since the laboratory test setup permitted very accurate control of the test conditions, it was felt that the five occasions of flame travel through the pump in an actual test proved conclusively that the laboratory test method truly represented the degree of safety which could be ex-

Table 2 – Results of Pump Explosion Tests

Spark Location	Tests Made	Mixtures Tried, %	Result
1	31	2 to 10	One external explosion at 4.55% mixture[a]
1a	6	4.5 to 8.7	No external explosion
1a, 2	8	7.6 to 9.3	One external explosion at 7.6% mixture[b]
1a, 3	6	7.6 to 9.1	One external explosion at 7.6% mixture[b]
1a, 4	8	7.6 to 9.4	Two external explosions at 7.6% mixture[a]
1, 4	9	4.5 to 8.7	No external explosion

[a] Internal explosion pressure, 2 psi.
[b] Internal explosion pressure, 10 psi.

Table 3 – Flame-Suppression Characteristics of Holes of Various Dimension – Pump Explosion Tests

Vent-Hole Size Checked		
Diameter	Length	Result
0.0810	2.06	Safe
0.0995	2.06	Unsafe
0.1100	2.06	Unsafe
0.1200	2.06	Unsafe
0.1250[a]	2.06	Unsafe
0.0625	0.68	Safe
0.0781	0.68	Unsafe
0.0939[a]	0.68	Unsafe
0.0465	0.25	Safe
0.0625	0.25	Unsafe
0.0783[a]	0.25	Unsafe
0.0465	0.18	Safe

[a] Diameter of actual holes in pump.

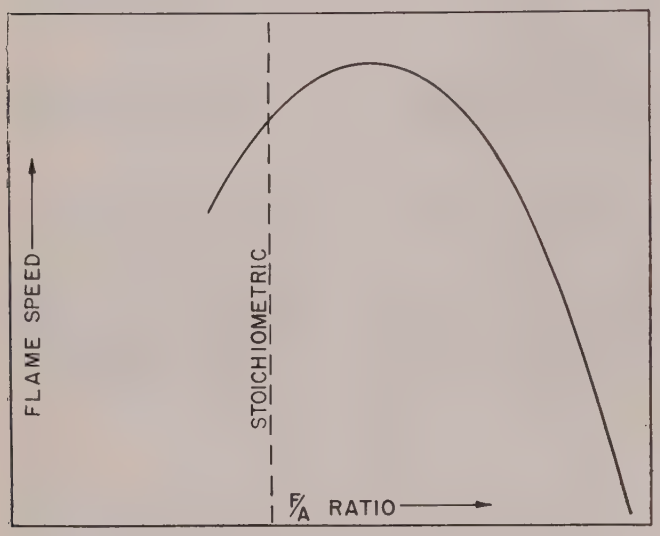


Fig. 5 – Typical curve showing flame speed of hydrocarbon mixture versus mixture ratios

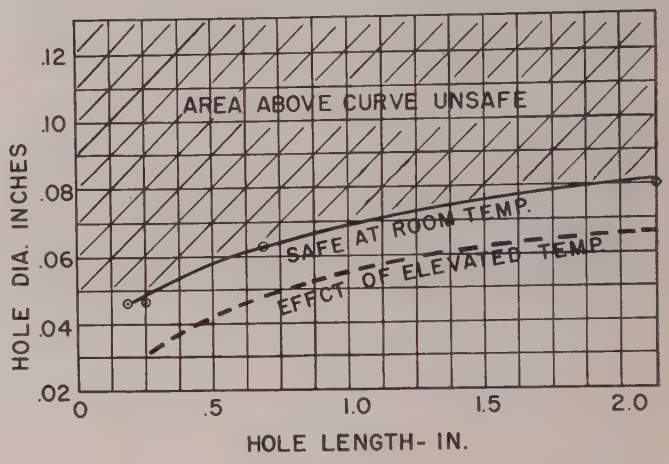


Fig. 6 – Effect of temperature on flame suppression through a small hole

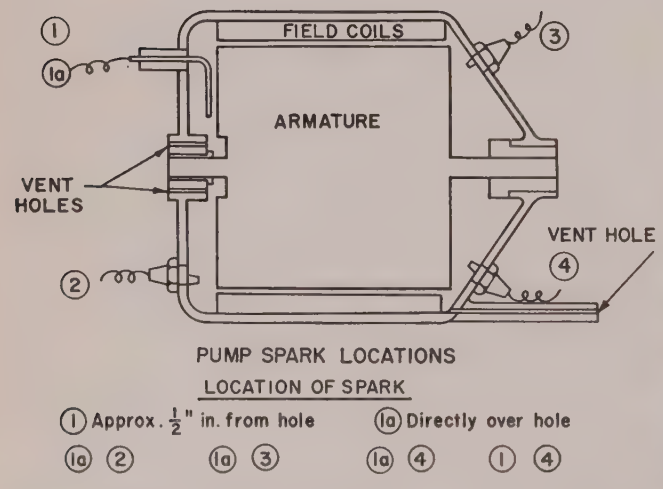


Fig. 7 – Schematic diagram showing spark locations for pump explosion tests

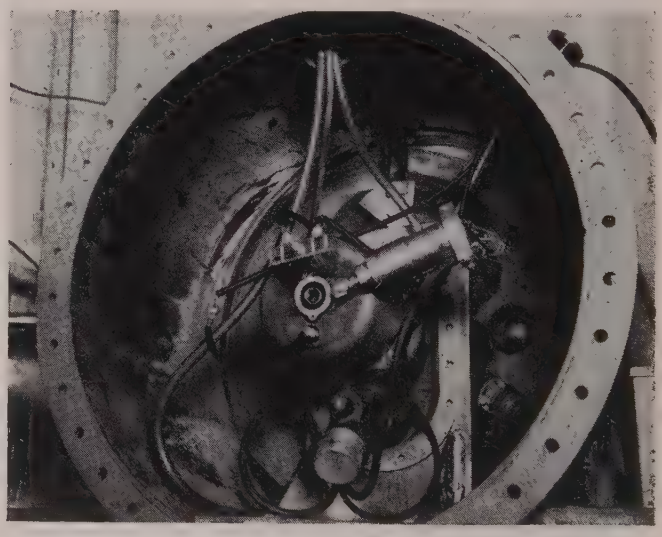


Fig. 8 – Test setup for pump explosion tests

pected from the hole size and length relationships used in the pump.

The difficulty to attain ideal explosive conditions in the actual pump test is exemplified by the high fuel/air ratios required to produce maximum pressures in the pump. Fuel/air ratios as high as this in the test bomb did not produce pressures as high or cause as great a flame propagation as did the lower mixture ratios.

Table 3 shows the comparison of pump vent holes which, from test results, were established as safe against those which were originally in the pump.

Conclusion

As a result of this investigation and considerable additional work on other types of flame-suppressing devices, Boeing established a new procedure for explosion-proof qualification for most of its aircraft equipment. This procedure is considerably more detailed than the normal procedures used for explosion testing. The test steps are such that the effects of such items as mixture ratio, ignition location, and temperature are reflected in the test results.

To the equipment manufacturer, our new procedure may appear unnecessarily severe. However, the requirements are such as to permit him to approach the design of new equipment as "designed safe" prior to actual fabrication. This is accomplished by a procedure in which the designer is given detail methods whereby he can proof test vent configurations in advance of final equipment design.

Experience to date has been that necessary changes to existing equipment for qualification under this procedure have been slight. In most cases it has been found that equipment manufacturers were pleased to find there was a way to approach equipment on a "designed safe" basis.

ORAL DISCUSSION

Reported by H. J. Hayden
Boeing Airplane Co.

F. A. Jennings, North American Aviation, Inc.: Did ignition ever occur in cases where no visible flame but only hot exhaust gases passed through the hole?

Mr. Klose: No evidence of such ignition was found during our testing.

Mr. Jennings: Were multiple small holes satisfactory as flame suppressors?

Mr. Klose: Yes, multiple small holes work satisfactorily.

Mr. Jennings: How did slits function as a suppressor as compared to holes?

Mr. Klose: Our work did not include tests with slits, however, from Bureau of Mines work one would guess that safe slits widths would be similar to safe hole diameters.

Question: Did conductivity of material affect flame propagation?

Mr. Klose: We did not test for this, but the literature shows test results identical for glass as for copper. Conductivity of material therefore apparently affects flame travel very little.

R. E. Gorton, Pratt & Whitney Aircraft: Is there any reason for the apparent importance of the location of ignition source with respect to flame-suppressor holes?

Mr. Klose: Literature on miscellaneous combustion studies led us to believe that it must be caused by the fact that gas molecules in the immediate vicinity of the point of ignition attain considerably higher temperature than those removed from the ignition source.

Mr. Gorton: Do you have any explanation as to why material conductivity is not important?

Mr. Klose: Theorists and other investigators claim it is a factor in screen suppressors but the element of time involved is so short an interval that the effect is inconsequential in comparison to other factors which affect flame travel through holes, which have an appreciable length in comparison to their diameter.

M. Belzer, North American Aviation, Inc.: Is there any effect of baffles in the chamber?

Mr. Klose: We have no information on this.

F. E. Lenherr, Northrop Aircraft, Inc.: Can quench-distance theory predict hole size?

Mr. Klose: Our understanding of this theory is limited. We found ourselves at a loss as to a means to include the effect of pressure drop across the opening since the theory was evolved to predict quenching under a constant-pressure burning condition not typical of aircraft applications.

Effect of Lubricant Composition

THE problem of combustion-chamber deposit formation in automotive engines has become increasingly important in recent years as gasoline engines of higher compression ratio and generally higher initial octane-number requirement have been introduced. It is well known that cars operated in normal driving increase from 5 to 15 numbers in octane requirement in a few thousand miles because of the formation of combustion-chamber deposits. If a method could be found to eliminate or minimize these deposits, it would be equivalent to a considerable boost in the octane number of present-day fuels. This would permit increasing the compression ratios of new cars even with current octane quality gasolines.

At the present time, the problem is being actively studied by both the petroleum and automotive industries, and in the last several years there have been a large number of technical papers devoted to this subject. The primary reason for the present interest is the increasingly high cost of producing higher octane quality gasolines to meet the anti-

MOLECULAR weight or volatility has been found to be the most important factor in determining the extent to which a lubricant contributes to combustion-chamber deposits and octane requirement. Bright stock appears to be particularly harmful. Crude source (or variation in predominating hydrocarbon type) seems to have little effect.

These authors report further that commercially practical mineral-base oils have been developed that show a significant advantage over previously known conventional lubricants in their ability to prevent these deposits and octane requirement increase when used with a conventional leaded fuel. Moreover, they say that SAE 5W-20 motor oils can be made that give a 4-5-octane advantage over conventional motor oils in preventing detonation.

They point out, further, that, although progress has been made, even better lubricants and fuels are needed, in this respect.

The Authors

J. G. McNab (M '44) is assistant director of the Esso Laboratories, Standard Oil Development Co. After receiving his Ph.D. in chemistry from the University of Chicago in 1936, he joined Esso Laboratories, where his work has been concerned largely with studies in the fields of fuels, lubricants, and additives.

L. E. MOODY is a research chemist with the Standard Oil Development Co. He received a B.A. degree from Wesleyan University. Upon completing his graduate work at the University of Wisconsin, he received a Ph.D. in physical chemistry.

N. V. HAKALA (M '47) is assistant director of the Esso Laboratories, Standard Oil Development Co. He attended Michigan College of Mining and Technology and received a B.S. degree in chemical engineering. He has been affiliated with the Esso Laboratories since 1943, following completion of graduate work leading to a Ph.D. in chemistry from the University of Wisconsin.

on Combustion-Chamber Deposits

J. G. McNab, L. E. Moody and N. V. Hakala, Esso Laboratories

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

knock requirements of modern high compression and more efficient engines. The catalytic cracking processes developed in the late Thirties not only resulted in increased volumes of gasoline but also gave high-octane-number gasoline components. Thus, the quality, as well as the volume, available to the motorist was increased. This permitted the automotive industry to raise the compression ratios of engines to take advantage of the higher-octane-number quality available.

Now the refiner is faced with producing even higher antiknock quality gasoline to meet the demand of new engines. To do this he is usually faced with the prospect of upgrading low-octane gasoline components by catalytic reforming processes such as hydroforming and platforming. Unlike catalytic cracking, these processes start with already existing gasoline fractions. It is no longer a question of converting less-valuable high-boiling components of petroleum for which the demand is relatively low into gasoline for which the demand is high. Now it is a question of processing material that is already gasoline, with increased octane quality as the objective.

Because this added processing involves some increased cost, the refiner is looking for all possible supplementary means to help satisfy the octane demand of present and future cars. One such alternative is the development of fuel and lubricant compositions that would minimize combustion-chamber deposit formation, thus resulting in lower ultimate engine octane requirements. The development of such products has been under study for several years by our laboratories. This paper reviews a portion of that work, with primary emphasis on the lubricant factors affecting combustion-chamber deposit formation.

Origin of Combustion-Chamber Deposits

To establish a firm basis for studying the problem, early work was directed toward determining conclusively the source of the combustion-chamber deposits. Did they come from the fuel, or the lubricant, or both? The results of both laboratory and field studies indicated clearly that both the fuel and the lubricant may contribute to deposit formation. The pertinent data leading to this conclusion are reviewed in the following paragraphs.

A. *Laboratory Studies on Origin of Deposits* – Data obtained early in laboratory studies showed that either the fuel or the lubricant could cause combustion-chamber deposits and an increase in octane requirement. In some cases, the fuels seemed to be the primary offender. For example, in cyclic multicylinder engine tests with a typical commercial gasoline, a 13-unit increase in octane requirement was observed with both a high-quality SAE 10 mineral oil X and an oxygen-containing synthetic oil Y. When the fuel was changed to isooctane, both of these oils again performed similarly. However, octane requirement increases of only 2-5 units were observed with the isooctane fuel. It appeared from these results that the commercial fuel alone might be the source of the deposits. Data illustrating these points are shown in Table 1. The test conditions used are given in Table 2. Inspection data on the lubricants used are given in Table 3.

In other cases, the lubricant seemed to be the primary offender in causing octane requirement

Table 1 – Effect of Fuel and Lubricant on Octane Requirement Increase

(Laboratory multicylinder engine; cyclic test[a])

	Octane Requirement Increase	
	SAE 10 Mineral Oil X	Synthetic Motor Oil Y
Commercial (1.5 Cc Tel) Fuel	13–18	13
Isooctane	5	2–4

[a] Test conditions are given in Table 2.

Table 2 – Laboratory Engine Test Conditions

I. Cyclic Test

L-head: 6-cyl engine
Cycle: ½ hr at idle
3 hr at 1700 rpm; 10 bhp
2½ hr at 2400 rpm; 20 bhp

Crankcase Oil Temperature, F	165
Jacket Temperature, F	165
Air Intake Temperature, F	110

II. Constant-Speed Test

Condition: Constant speed at 2400 rpm; 20 bhp

Crankcase Oil Temperature, F	165
Jacket Temperature, F	165
Air Intake Temperature, F	110

Table 3 – Lubricant Inspections

	Solvent-Extracted Midcontinent SAE 10 (Oil X)	Oxygen-Containing Synthetic (Oil Y)
Gravity, deg API	31.1	14.9
Conradson Carbon, % by weight	0.10	0.04
Viscosity at 100 F, SUS	192	180
Viscosity at 210 F, SUS	46.7	48.6
Viscosity Index	109	138
Ash, % by weight	Trace	Nil

Table 4 – Effect of Fuel and Lubricant on Octane Requirement Increase

(Laboratory multicylinder engine; constant-speed test[a])

	Octane Requirement Increase	
	SAE 10 Mineral Oil X	Synthetic Motor Oil Y
Commercial (1.5 Cc Tel) Fuel	8	5
Hydrogen	10	—

[a] Test conditions are given in Table 2.

Table 5 – Effect of Fuel and Lubricant Composition on Octane Requirement Increase

(Controlled field test; synthetic lubricant Y)

	Octane Requirement			Deposits (Per Engine)	
	Initial	Equilibrium	Increase	Weight, g	Volume, cc
Commercial Gasoline	77	87	10	56.8	17.6
Isooctane	79	78	−1	2.3	0.9

increase. In one set of laboratory engine tests carried out with the same high-quality SAE 10 straight mineral oil discussed above there was little difference in the octane requirement increase obtained with fuels as widely different as hydrogen and a commercial gasoline (1.5 cc tel per gal). In the hydrogen fuel experiment, the deposits must have come from the lubricant. This experiment demonstrated that, at least under the test conditions employed, the lubricant was the source of deposits that caused a large octane requirement increase. This is shown in Table 4. The test conditions used are given in Table 2.

The data in Tables 1 and 4 indicate that combustion-chamber deposits may come from either the fuel or the lubricant, or perhaps both. Similar data illustrating the point that the deposits may come from either the fuel or the lubricant may be found in recent publications. In one paper[1] considerable difference in octane increase in laboratory tests was shown between a commercial leaded gasoline and an unleaded aviation gasoline consisting mostly of alkylate and virgin stocks. Little difference was shown among lubricants ranging from a straight mineral oil to a Supplement 2 detergent oil. Limited data[2] from a field test carried out in the Los Angeles area also indicated that there was little difference in performance between a conventional mineral oil and an ester-type synthetic motor oil when evaluated with leaded commercial gasolines. Both combinations gave appreciable octane requirement increase. With isooctane and the synthetic oil, very good performance, as indicated by only a 2-unit octane requirement increase, was obtained. Unfortunately, data were not reported on the mineral oil run with isooctane.

B. *Field Studies on Origin of Deposits* – Extensive laboratory data from the authors' studies and the available published data both pointed up the need for controlled field studies to clarify the role of fuel and lubricant in combustion-chamber deposit formation under actual service conditions. To obtain the needed data, we undertook an extensive experimental field program, using the facilities of the Southwest Research Institute at San Antonio, Tex. This program was carried out using sixteen 1951 cars equipped with overhead-valve, V-type engines of 7.5/1 compression ratio. After cleaning the combustion chambers, these cars were operated caravan-style on the test fuels and lubricants under controlled low-speed urban and suburban driving conditions. Octane requirements were

[1] "Engine Laboratory Study of Motor Oil Characteristics as One of Factors Influencing Combustion-Chamber Deposits," by D. E. Carr, L. A. McReynolds, S. C. Britton, and R. E. Linnard. Presented at the SAE Annual Meeting, Detroit, Jan. 17, 1952.

[2] "Effects of Combustion-Chamber Deposits on Knock," by W. E. Bettoney. Presented before the SAE Metropolitan Section, New York City, April 3, 1952.

measured by the standard Uniontown procedure. The results are reported as equilibrium octane requirements after deposit buildup, rather than octane requirement increase, because the equilibrium requirement data are more reproducible and also represent what the average driver sees. Each fuel-lubricant combination was run in two cars. The repeatability of single determinations using 2 cars (2-car average) is believed to be in the order of ±2 octane numbers. Details of the controlled field test procedure and data showing the reproducibility of the test are given in the Appendix.

The data from the controlled field test definitely showed that combustion-chamber deposits may come from either the fuel or the lubricant, or both. As shown in Table 5 and Fig. 1, when the oxygen-containing synthetic oil Y was used with isooctane, there was no increase in octane-number requirement, indicating that neither isooctane nor the synthetic oil Y contributes to combustion-chamber deposits. With both a noncontributing fuel and lubricant clearly established, it was then possible to measure the deposit-forming tendencies of other fuel and lubricant compositions. When the synthetic oil Y was used with a commercial gasoline A (about 2 cc tel per gal) the equilibrium requirement was 87, or about 10 octane numbers above the initial. In this case the deposits must have come from the fuel. Inspections on the commercial gasoline A, in comparison with isooctane, are given in Table 6.

When the SAE 10 straight mineral oil X was used with isooctane, an equilibrium requirement of 90 was obtained. Since isooctane is not a contributor, the deposits in this case must have come from the lubricant. When the commercial gasoline A and the SAE 10 mineral oil X were used together, the octane requirement (91) was essentially the same as with isooctane and the mineral oil (90). However, it was significantly higher than with the commercial fuel and the synthetic oil (87). In the case of the commercial gasoline, mineral oil X combination, the deposits probably came from both the fuel and the lubricant. These data, summarized in Table 7, help explain many of the apparent discrepancies seen in the past. The results obtained previously have frequently been from a combination fuel and lubricant effect, so that clearcut conclusions regarding the specific influence of either the fuel or the lubricant component were not possible.

The data in Table 7 indicating that there are significant differences between lubricants with a leaded commercial gasoline are in good agreement with data reported by du Pont. As shown in Table 8, in the du Pont tests[3] with a leaded commercial

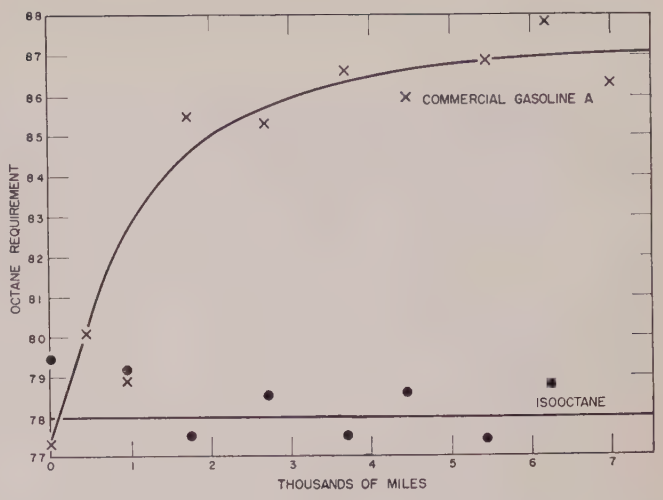


Fig. 1 – Effect of fuel and lubricant on octane increase – controlled field test; synthetic lubricant Y

Table 6 – Fuel Inspections

	Commercial Gasoline A	Isooctane
Tel, cc per gal	1.90	Nil
Gravity, deg API	64.0	73.1
Reid Vapor Pressure, psi	8.7	1.8
Research Octane Number	94.6	100
Bromine Number, cg per cc	48.3	—
% Olefins	40	Nil
% Aromatics	11	Nil
Distillation (ASTM):		
Ibp, F	100	205
Fbp, F	372	264
5% Point, F	122	208
50% Point, F	201	210
90% Point, F	309	216
95% Point, F	338	221

Table 7 – Effect of Fuel and Lubricant Composition on Octane Requirement

(Controlled field test results)

	Equilibrium Octane Requirement[a]	
	SAE 10 Mineral Oil X	Synthetic Motor Oil Y
Commercial Leaded Gasoline A	91	87
Isooctane	90	78

[a] Average initial requirement was 78.

Table 8 – Effect of Fuel and Lubricant Composition on Combustion-Chamber Deposits

(Average of two cars on each fuel-lubricant combination)

	Equilibrium Octane Requirements			
	du Pont Data		Esso Laboratories Data	
	Commercial Leaded Gasoline	Isooctane	Commercial Leaded Gasoline A	Isooctane
Mineral Oil	88	—	91	90
Synthetic Oil	85	76	87	78

[3] "Combustion-Chamber Deposits and Their Effects on Engine Fuel Requirements," by J. J. Mikita and W. E. Bettoney. Presented before Western Petroleum Refiners Association, San Antonio, Tex., March 23-25, 1953.

Table 9 – Origin of Deposits – Comparison of Data from Dual Fuel System Car and Controlled Field Test

(Synthetic lubricant Y)

	Octane Requirement Increase		Combustion-Chamber Deposits, g	
	Common Crankcase	Separate Engines[a]	Common Crankcase	Separate Engines[b]
Commercial Leaded Gasoline A	6	10	22	29
Isooctane	1	0	2	1

[a] 2-car average.
[b] Corrected to 4 cylinders.

gasoline there was about a 3-octane-number difference between a mineral oil and an ester-type synthetic motor oil. An additional 9-octane-number advantage was obtained by switching from the commercial leaded gasoline, synthetic motor oil combination to a combination of isooctane fuel and the synthetic ester lubricant.

The significant differences indicated in the equilibrium octane requirements observed with different oils suggested that the influence of oil composition on combustion-chamber deposit formation would be a desirable field to investigate. Work along these lines is discussed in a subsequent section of this paper.

C. *Mechanism of Deposit Formation* – Considerable thought and study have been devoted at our laboratories and elsewhere to determining what the mechanism of combustion-chamber deposit formation is and how the deposits exert their harmful effect. There have been a number of excellent papers[4, 5] on these subjects, particularly with regard to how the deposits exert their effect. There seems to be general agreement that the deposits act by exerting (a) a volume effect, in that they increase the effective compression ratio, and (b) a thermal effect, which may be due either to the thermal-insulating or perhaps the heat-capacity characteristics of the deposits. It has also been suggested that the deposits may have a catalytic effect, but the evidence for this is less clear cut.

Our studies largely confirm the work already reported in the literature[4] showing that the thermal effects are probably most important. The volume, or compression ratio, effect appears to account for only 15-20% of the deposit harm. There is no direct evidence of a catalytic effect. However, the probability exists that catalytic effects are important, in that the thermal and volume effects do not always seem to account for all the observed results. Inasmuch as these results are in general agreement with published data, they will not be reviewed further here.

We have also carried out some experimental work to determine where, or how, the deposits, which have been shown to come from both the fuel and the lubricant, are formed. In other words, are they formed directly in the combustion chamber or do blowby products from the fuel react in, or with, the lubricant to form resins that are later laid down in the combustion chamber? To provide information on this point a car with a General Motors Research engine (8/1 compression ratio) was equipped with a dual fuel system, which fed four cylinders on each of two fuels. An experiment was carried out in this car using the previously described commercial gasoline A containing about 2 cc of tel per gal in four of the cylinders and isooctane in the other four. The synthetic lubricant Y was used as the common crankcase lubricant. The synthetic oil has been shown to be a noncontributor, so that there should be no effect due to the lubricant. The data indicate that the deposits are formed directly in the combustion chamber. After approximately 4000 miles of city-country driving in the Elizabeth, N. J., area, the four cylinders operating on the leaded commercial gasoline had increased about 6 octane numbers in requirement. The four cylinders operating on isooctane had increased only 1 octane number in requirement. A comparison of the data obtained in the dual fuel system car and data obtained in the controlled field tests described previously is shown in Table 9. In this table the data are reported on an octane requirement increase basis because of the large difference in the initial octane requirements of the GM Research engine and the passenger cars used in the field studies, due to the higher compression ratio (8/1) of the GM engine.

In general, the agreement of the octane requirement increase and also the deposit weights between the dual fuel system car and the separate cars is very good, as shown in Table 9.

These data indicate that in the dual fuel system car the deposits were formed directly in the combustion chamber and that deterioration products resulting from crankcase dilution or deposits arising from blowby gases were not a factor. If, for example, lead salts accumulated in the crankcase oil from the leaded fuel reached the combustion chambers as oil was consumed, the deposits from the cylinders run on isooctane should have contained lead. Only trace amounts of lead were found in the deposits from these cylinders, indicating that the deposits must have been formed directly in the combustion chamber. It is, of course, possible that, under other conditions, such as with

[4] See SAE *Quarterly Transactions*, Vol. 5, October, 1951, pp. 565-576: "Possible Mechanisms by Which Combustion-Chamber Deposits Accumulate and Influence Knock," by L. F. Dumont.

[5] See SAE Transactions, Vol. 61, 1953, pp. 361-377: "Combustion-Chamber Deposition and Knock," by H. J. Gibson, C. A. Hall, and D. A. Hirschler.

other lubricants, fuel decomposition products in the oil may play some part in the formation of combustion-chamber deposits.

Lubricant Factors Affecting Combustion-Chamber Deposits

Since the deposits come from both the fuel and the lubricant, it is, of course, necessary to have both a noncontributing fuel and a noncontributing lubricant really to solve the combustion-chamber deposit problem. Much effort has been devoted to studying the factors involved in developing noncontributing fuels and lubricants. Considerable progress has been made in isolating the lubricant factors causing combustion-chamber deposits. These are discussed in greater detail in the following sections of this paper. The fuel factors affecting deposit formation are not discussed here.

It was shown previously that some oxygen-containing synthetic oils cause little or no combustion-chamber deposits. However, these oils are generally very high in cost and of limited availability. In many instances they may also have shortcomings with regard to providing overall satisfactory lubrication of automotive engines. On the other hand, petroleum-based lubricants having outstanding overall performance characteristics can be formulated in large volume and at relatively low cost. For these reasons, much of our laboratory effort has been devoted to developing mineral oils having low combustion-chamber deposit-forming tendencies.

In this work the effect of the more important lubricant variables on deposit formation and octane requirement increase has been studied using the controlled field test procedure described previously and given in detail in the Appendix. The variables studied included the following:

1. Presence of residual materials (bright stock).
2. Neutral distillate volatility.
3. Base stock type.
4. Polymeric thickening agent type.
5. Detergent-inhibitor type and concentration.
6. Effect of oil oxidation.
7. Use of "combustion" catalysts in the lubricant.

Based on the results of this work, a commercial lubricant has been formulated having low combustion-chamber deposit and octane requirement increase characteristics, as well as otherwise satisfactory overall performance properties. Uncontrolled field tests and service experience have confirmed the results obtained in this study.

A. *Studies Using Isooctane as Fuel* – Much of the work to study the effect of lubricant variables was carried out with isooctane as fuel to eliminate any contribution of the fuel to the deposits. The trends observed using this fuel have been confirmed in studies with commercial gasolines, so that the use of a noncontributing fuel like isooctane represents a practical means of isolating the lubricant variables.

Table 10 – Effect of Bright Stock on Combustion-Chamber Deposits

(Controlled field tests; isooctane fuel)

% Bright Stock in Solvent-Extracted Midcontinent Distillate	Equilibrium Octane Requirement[a]	Combustion-Chamber Deposits	
		Weight, g	Volume, cc
0	84	11.4	6.9
8	90	27.0	22.5

[a] Average initial requirement 78.

1. Effect of residual material (bright stock): The highest molecular weight and least volatile component of a lubricating oil base is bright stock. The effect of this type of material on combustion-chamber deposits was determined using a blend made by adding 8% of a solvent-extracted, deasphalted, dewaxed, Midcontinent residuum of 165 SUS viscosity at 210 F, to a distillate having a viscosity of 44 SUS at 210 F. The distillate used was also solvent extracted and dewaxed and came from a Midcontinent crude source. The viscosity of the resulting blend was about 48 SUS at 210 F. Both the blend and the distillate were evaluated in the controlled field test cars using isooctane as fuel. The lubricant blend containing the bright stock gave much larger amounts of combustion-chamber deposits and a 6-unit higher octane-number requirement than was obtained with the straight distillate. This is illustrated in Table 10.

These data in Table 10 indicate quite clearly that to produce a noncontributing lubricant, heavy components, such as the residuum type of material, should be eliminated from the base oil.

2. Effect of distillate volatility: Another important factor was considered to be the presence of heavy components in the neutral distillates used in the lubricant blend. The effect of eliminating the heavier components from the distillate used as the base case in the previous experiment was investigated.

For this purpose, a narrow-cut, lower-boiling distillate was prepared from the same refinery streams used to produce the base case distillate. The viscosity of the narrow-boiling distillate was 65 SUS at 100 F, compared to 165 SUS for the base case distillate. To provide adequate viscosity to avoid excessive oil consumption, the lower-viscosity distillate was thickened with a viscosity index improving olefin polymer (V.I. improver M) to a viscosity of 48.8 SUS at 210 F. This compares to about 44 SUS at 210 F for the base case wide-cut distillate. These oils were then evaluated in two cars, each using isooctane fuel in the controlled field test. The narrow-cut product gave no increase

Table 11 – Effect of Lubricant Volatility on Combustion-Chamber Deposits

(Controlled field tests; isooctane fuel)

Solvent-Extracted Distillate	Viscosity at 100 F, SUS	Equilibrium Octane Requirement[a]	Combustion-Chamber Deposits	
			Weight, g	Volume, cc
Wide Cut	165	84	11.4	6.9
Narrow Cut[b]	65	76	4.8	3.7

[a] Average initial requirement 78.
[b] Contained V.I. improver M as thickening agent to give 48.8 SUS at 210 F, versus 44 SUS for wide-cut distillate.

Table 12 – 10-Mm Vacuum Engler Distillation Test (ASTM-D 1160-52T)

	Viscosity at 100 F, SUS	Distillation, F					
		Ibp	10%	50%	90%	Fbp	Δfbp–ibp]
Narrow-Cut Distillate	65	325	380	432	479	508	183
Wide-Cut Distillate	165	370	428	511	614	660	290

Table 13 – Effect of Crude Type on Combustion-Chamber Deposits (Controlled field tests; isooctane fuel)

Solvent-Extracted Distillates	V.I.	Viscosity at 210 F, SUS	Equilibrium Octane Requirement[a]	Combustion-Chamber Deposits	
				Weight, g	Volume, cc
Paraffinic	112	44	84	11.4	6.9
Naphthenic	40	55	83	7.3	6.2

[a] Average initial requirement 78.

Table 14 – Inspection Data for Paraffinic and Naphthenic Distillates

	Paraffinic Distillate	Naphthenic Distillate
Viscosity, SUS		
At 100 F	165	513
At 210 F	44	55
Viscosity Index	112	40
% Paraffin Chains	69	48
% Naphthene Rings	27	42
% Aromatic Rings	4	10
10-Mm Vacuum Distillation,[a] F		
Ibp	370	360
50%	511	534
90%	614	611
Fbp	660	645

[a] ASTM – D1160-52T.

Table 15 – Effect of Crude Type on Combustion-Chamber Deposits (Controlled field tests; isooctane fuel)

Lubricant Type[a]	Base Stock Inspections		Equilibrium Octane Requirement[b]	Combustion-Chamber Deposits	
	V.I.	Viscosity at 210 F, SUS		Weight, g	Volume, cc
Paraffinic	105	36	83	16.1	9.8
Naphthenic	40	40	83	17.6	9.2

[a] Contain viscosity index improver M and 5% detergent-inhibitor.
[b] Average initial requirement about 78.

in octane requirement after 6000 miles, with an equilibrium octane requirement of 76. This compared to an octane requirement of 84 obtained with the base case wide-cut oil, as shown in Table 11. This difference in requirement is attributed to the removal of the heavy ends in the lower-boiling lubricating oil base stock. This removal of heavy ends is also reflected in the lower amount of combustion-chamber deposits found with the narrow-cut oil (Table 11).

The octane requirement data of Table 11 are compared with the previous data on the effect of bright stock in Fig. 2. The adverse effect of high-molecular-weight materials is quite conclusively shown in this figure. The extent to which heavy materials were actually removed in the narrow-cut distillate is shown by the data in Table 12 from a 10-mm vacuum Engler distillation test.

The vacuum Engler distillation data for these same two distillates are also shown in Fig. 3, in comparison with the data for the bright stock blend discussed previously. It will be noted that the narrow-cut distillate had a considerably lower final boiling point than the wide-cut distillate. Also, cracking was encountered with the bright stock blend, indicating the presence of very heavy material, whereas no cracking was observed with the two distillate oils.

The data discussed above indicate that the first step in preparing a lubricant that does not contribute to combustion-chamber deposit formation is to start with a low-molecular-weight distillate of controlled volatility as the base oil.

3. Effect of base stock type: It has been claimed frequently over the years that naphthenic oils have less tendency to give combustion-chamber deposits than paraffinic oils. However, there does not appear to be a significant difference between paraffinic and naphthenic oils of the same volatility characteristics based on data obtained in this study. The crude source and variations in the predominating types of hydrocarbon components in lubricating oil distillates appear to be relatively unimportant, compared to the factor of base stock volatility.

For example, an acid-treated naphthenic oil from Coastal crude was compared in the controlled field tests with the wide-cut solvent-extracted paraffinic oil from the Midcontinent crude previously discussed. The two oils were very similar in volatility characteristics. The cars operated on these two distillate oils and isooctane fuel came to very nearly the same equilibrium octane requirement of 83-84. Data summarizing this performance are shown in Table 13.

Some of the pertinent inspections of these two distillates are given in Table 14. It will be noted that the two oils have quite different viscosity characteristics. While the naphthenic oil was much more viscous, it was specifically chosen because it closely matched the volatility of the paraffinic distillate. There is a significant difference in the

makeup of the oils from the standpoint of basic hydrocarbon types (ratios of paraffins to naphthenes to aromatics). If a major effect of crude type existed, it should have been observed in the preceding tests with these oils.

No effect of base stock type was observed in further studies with two finished low viscosity motor oils. These were also evaluated with isooctane as the fuel. One oil was based on a paraffinic distillate and the other on a naphthenic distillate. These distillates were of lower viscosity than those used in the experiments described above. Both oils contained viscosity index improver M and 5% of a commercial detergent-inhibitor additive. As shown in Table 15, no effect for crude source was noted either in the octane requirement levels reached or the amounts of deposits formed.

These data from Table 15 are shown in comparison with the previous data for the distillates in Fig. 4. It will be noted that more deposits were formed with the finished lubricants containing the detergent additives. However, for some unexplained reason, this did not seem to affect adversely the equilibrium octane requirement. These data on the absence of effect of crude type are in agreement with other data reported recently,[5] namely, that crude type was not as important as lubricant molecular weight or volatility in determining the extent of combustion-chamber deposit formation.

4. Effect of thickening agents (V.I. improvers): Once it has been established that it is possible to make noncontributing base stocks from narrow-cut low viscosity neutral distillates, the problem of producing a noncontributing finished lubricant is based on finding suitable additives to provide the other necessary properties in the oil, such as proper viscosity, detergency, and so on, but that do not themselves cause serious octane requirement increase.

The problem of providing adequate viscosity has been easily resolved. Since bright-stock and higher-molecular-weight fractions of distillates are major contributors, they are eliminated, and the thickening is accomplished by polymeric materials. However, careful selection of the thickening agent used is necessary, since not all types of products used for thickening appear to be satisfactory. The effect of adding a commonly used polyolefin-type thickening agent, V.I. improver M, in a narrow-cut solvent-extracted Midcontinent distillate is shown in Table 16. Because of its chemical structure this material would be expected to decompose on combustion and leave no carbonaceous residue. Past studies on certain special industrial lubricants containing this additive had, in fact, indicated this to be the case. It will be noted that the mineral oil blend of this material gave the same or lower equilibrium octane requirement as was obtained with the noncontributing synthetic oil Y. This indicates that V.I. improver M does not contribute to deposits.

Not all V.I. improvers are necessarily free of

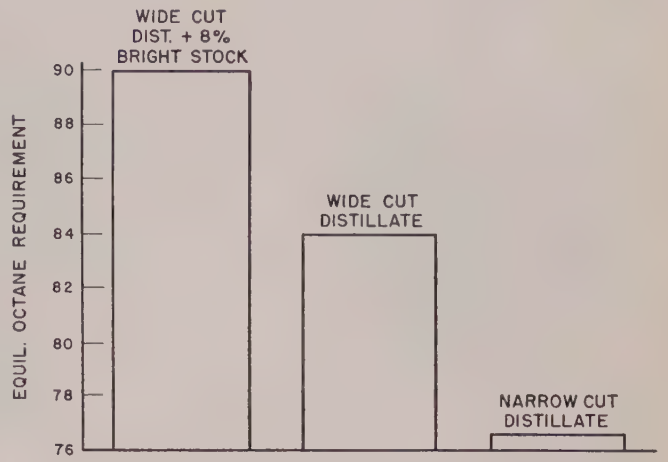


Fig. 2 – Effect of lubricant volatility on octane requirement – overhead-valve, V-type engines; 2-car averages; 6000 miles; isooctane fuel

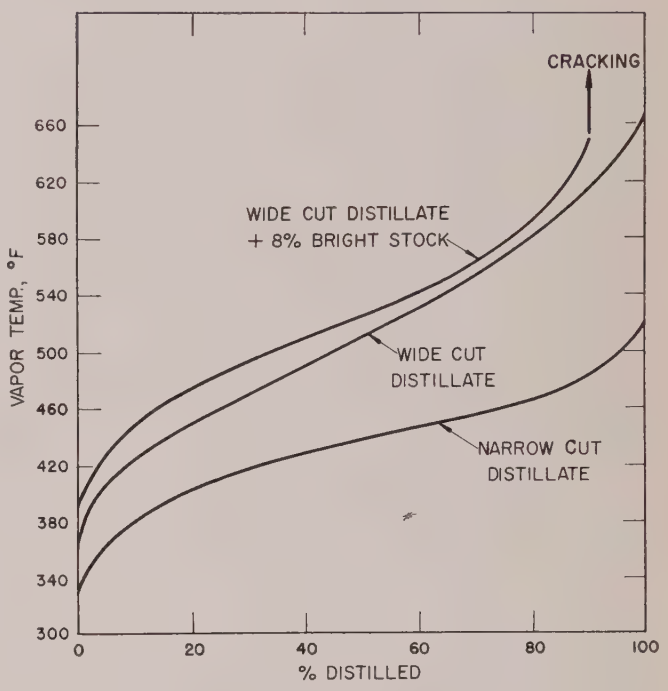


Fig. 3 – Volatility of lubricants – 10-mm vacuum Engler distillation test, ASTM-D 1160-52T

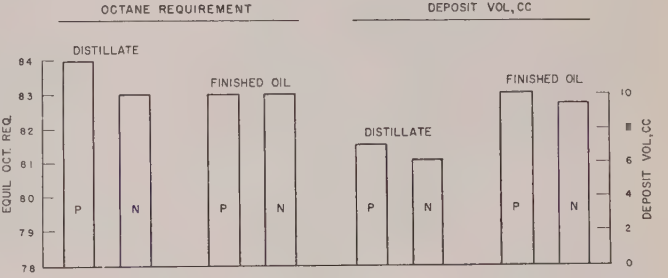


Fig. 4 – Effect of lubricant type on octane requirement – overhead-valve, V-type engines; 2-car averages; 6000 miles – P = paraffinic, N = naphthenic

Fig. 5 – Type of deposits obtained with V.I. improvers – left: V.I. improver M; right: V.I. improver P

combustion-chamber deposit-forming tendencies. For example, laboratory engine test results on a commercial V.I. improver P, of quite different chemical composition than V.I. improver M, indicate that it does contribute to deposits and to octane requirement increase. Based on correlations of these laboratory engine test data with the controlled field test results, blends of this V.I. improver P in the low viscosity distillate would be expected to give about a 7-unit higher octane requirement in the field, when using a noncontributing fuel such as isooctane, than was found with the V.I. improver M blend.

It is believed that the physical nature of the deposits obtained with the oil blend containing V.I. improver P may be responsible for the higher octane requirement observed. These deposits appeared to form an adherent continuous film on the inside of the combustion chamber; the deposits from the blend with V.I. improver M are granular in nature and more readily removable. This is illustrated in Fig. 5. The adherent film obtained with the V. I. improver P blend would be expected to have high thermoinsulating properties. As pointed out by Dumont,[4] continuous films lead to high octane requirements, largely because of their good thermal-insulating properties.

Field test data on another polymeric additive, T, having a chemical structure similar to that of V.I. improver P, confirmed the fact that materials of this type form deposits that result in high octane requirements in actual service. Additive T has both V.I. improver and detergent properties. The increased octane requirement data are shown in Table 17.

Larger amounts of deposits were obtained with the blend containing additive T than with the blend containing V.I. improver M. In addition, a continuous, adherent deposit film was formed on the piston top and cylinder head with additive T. It is believed that the physical characteristics of this film account for the high requirements observed. A comparison of the deposits obtained with the blends of additive T and V.I. improver P is shown in Fig. 6. The continuous film-like nature of the deposits from both of these materials is readily apparent. These data confirm the fact that careful selection of additives used in lubricating oils is

Table 16 – Effect of Viscosity Index Improvers on Combustion-Chamber Deposits
(Controlled field tests; isooctane fuel)

	Viscosity of Blend at 210 F, SUS	Equilibrium Octane Requirement	Deposits	
			Weight, g	Volume, cc
2.6% V.I. Improver M in Low Viscosity Distillate	48.8	76	4.8	3.7
Synthetic Oil Y	48.6	78	2.3	0.9

Table 17 – Effect of Additives on Combustion-Chamber Deposits
(Controlled field test; isooctane fuel)

% Additive in Low Viscosity Solvent-Extracted Midcontinent Base Stock	Equilibrium Octane Requirement[a]	Combustion-Chamber Deposits	
		Weight, g	Volume, cc
2.6% additive M	76	4.8	3.7
5% additive T	86	8.6	5.9

[a] Average initial requirement about 78.

required to avoid excessive combustion-chamber deposit formation and octane requirement increase.

5. Effect of detergent-inhibitor additives: The problem of finding a completely noncontributing detergent-inhibitor additive has yet to be resolved. However, by proper selection and by the use of suitable low viscosity base stocks of controlled volatility, detergent oils giving low octane requirements can be formulated.

Detergent-inhibitor additives are commonly used in passenger-car motor oils in concentrations varying from 1 to about 10%, depending on the level of detergency desired. A number of detergent-inhibitor additives have been evaluated to date in these studies, and all of them have contributed to some extent to combustion-chamber deposits. However, the present field studies have shown that it is possible to make finished oils giving relatively low increases in octane requirement, even though the detergent-additive may contribute to some extent to the formation of combustion-chamber deposits. For example, field tests were carried out on a finished lubricant based on the narrow-cut solvent-extracted Midcontinent distillate (65 SUS viscosity at 100 F) referred to previously. The finished oil contained 5% of detergent-inhibitor R, V.I. improver M, and a pour depressant. Detergent-inhibitor R is a well-known, widely used metal-containing detergent-inhibitor additive. With isooctane as fuel the oil blend containing these additives gave an equilibrium octane requirement in the controlled field tests of 83. This compares with an equilibrium octane requirement of 90 obtained under the same conditions with mineral oil A described previously. In this case, detergent-inhibitor R actually added about 7 octane numbers to the requirement (76) obtained with the straight narrow-cut distillate plus V.I. improver M. Even so, the performance of the oil containing the detergent-inhibitor is superior to that of the straight SAE 10 mineral oil A. The base case blend with no detergent gave no increase in octane requirement. This is shown in Table 18.

Table 18 – Effect of Detergent-Inhibitor Additives on Combustion-Chamber Deposits

(Controlled field tests; isooctane fuel)

Additive in Low Viscosity Solvent-Extracted Midcontinent Base[a]	New Oil Sulfated Ash, % by weight	Equilibrium Octane Requirement[b]	Combustion-Chamber Deposits	
			Weight, g	Volume, cc
None	0	76	4.8	3.7
5% detergent-inhibitor R	0.54	83	16.1	9.8
Mineral oil A – no additive	0	90	27.0	22.5

[a] Containing V.I. improver M.
[b] Average initial requirement about 78.

The data in Table 18 illustrate that finished oils containing V.I. improver and detergent-inhibitor additives can be made that give lower octane requirements with a noncontributing fuel than were obtained with previously available types of mineral-based oils. As will be shown in a later section, improved oils of this type also have an advantage over previously used oils when using a leaded commercial gasoline.

In further experiments, it was found that the equilibrium octane requirement obtained with blends containing a higher concentration of detergent-inhibitor R increased markedly as additive concentration was increased. This is illustrated in Fig. 7 and Table 19. It will be noted that increasing the concentration of the detergent additive from 5 to 14% increased the equilibrium octane-number

Fig. 6 – Type of deposits obtained with polymeric ester-type materials – left: V.I. improver P; right: detergent T

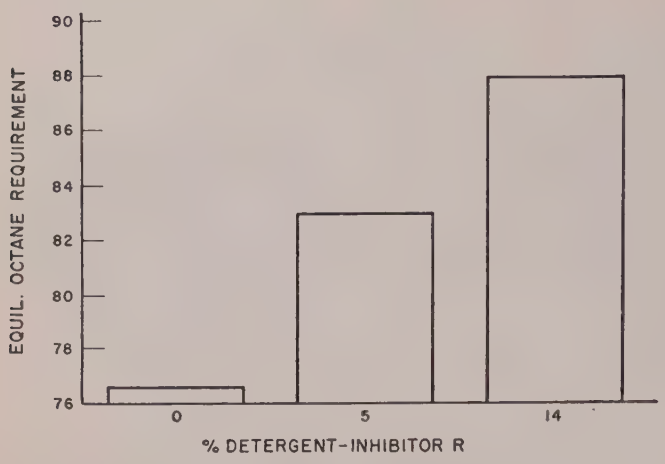


Fig. 7 – Effect of detergent-inhibitor on octane requirement – overhead-valve, V-type engines; 2-car averages; isooctane fuel – blends in low viscosity distillate containing V.I. improver M

requirement from 83 to 88. There was a corresponding increase in the amount of deposit formed.

The data in Table 19 suggest that an optimum balance exists between the minimum detergent additive level required for maintaining a satisfactory degree of cleanliness characteristics in a lubricating oil and the maximum that can be tolerated from a combustion-chamber deposit standpoint. Careful selection of the proper concentration and type of additive is required to obtain the best possible overall performance. It should be emphasized that the above data were obtained with a noncontributing fuel (isooctane). With a leaded commercial fuel, the adverse effect of increasing the detergent additive content might be less easily observed.

6. Effect of oil oxidation: One additional factor investigated briefly was the possible effect of oxidation products of the lubricant. In an early test with a naphthenic-base oil it was noted that the octane requirement level determined immediately following an oil change was lower than the requirement determined immediately prior to the oil change. This was not observed with the paraffinic-type oil. Data comparing the effect on octane requirement of oil changes with the naphthenic distillate as contrasted to the SAE 10 paraffinic oil are shown in Table 20.

Since naphthenic oils are known to be inherently unstable toward oxidation, it was felt that the high octane requirements observed just prior to oil changes might be caused by the formation of oil oxidation products, which were contributors to combustion-chamber deposits. It was considered possible, also, that if these oxidation products could be eliminated, essentially no octane requirement increase would be observed with this oil.

To throw light on the possible effect of oil oxidation, the naphthenic oil was re-evaluated, again using isooctane as the fuel. In this experiment the oil was redistilled prior to use to eliminate any oxidation products that might have built up in storage. To cut down on oxidation products developing in the oil during the course of the test, the oil was inhibited with a phenolic-type antioxidant. To further ensure that oxidation products did not build up in the oil during the test, the crankcase charge was changed every 1000 miles, instead of every 2000 miles. As illustrated in Table 21, this procedure eliminated the drop in octane requirement increase due to the oil change. However, it had no effect on either the equilibrium octane requirement or amount of deposits formed. There was some reduction in oxidation products, as illustrated by the data on increase in used oil viscosity.

The data in Table 21 indicate that oil oxidation products are not a major factor in the formation of combustion-chamber deposits.

7. Effect of combustion catalyst in lubricant: All additive in the lubricating oil that would remove combustion-chamber deposits or modify them so they do not cause harm would be the ideal solution to the problem. Considerable effort has been devoted to finding such an additive. Data reported by

Table 19 – Effect of Detergent-Inhibitor Additive Concentration on Combustion-Chamber Deposits

(Controlled field tests; isooctane fuel)

% Additive in Low Viscosity Solvent-Extracted Midcontinent Base Stock[a]	New Oil Sulfated Ash Content, % by weight	Equilibrium Octane Requirement[b]	Combustion-Chamber Deposits	
			Weight, g	Volume, cc
None	0	76	4.8	3.7
5% detergent-inhibitor R	0.54	83	16.1	9.8
14% detergent-inhibitor R	1.45	88	24.8	16.6

[a] Contains viscosity index improver M.
[b] Average initial requirement about 78.

Table 20 – Effect of Oil Change on Combustion-Chamber Deposits

(Controlled field tests; isooctane fuel)

	Octane Requirements Before and After Oil Changes			
	Naphthenic Oil		SAE 10 Solvent-Extracted Midcontinent Oil A	
Oil Change Point:	Before	After	Before	After
2200 Miles	81.5	80.0	79.3	82.0
4400 Miles	85.2	81.3	88.0	87.3
6600 Miles	82.8	81.3	89.8	90.5
Average	83.2	80.9	85.5	86.6
Average Change in Requirement	−2.3		+1.1	

Table 21 – Effect of Oil Oxidation on Combustion-Chamber Deposits
(Controlled field test; isooctane fuel; naphthenic oil)

% Oxidation Inhibitor	0	4
Oil Drain Period, miles	2000	1000
Average Change in Requirement Before and After Oil Change	−2.3	+0.9
Equilibrium Octane Requirement	83	84
Combustion-Chamber Deposit, g	7.3	8.3
Lubricant Viscosity at 100 F, SUS		
New	513	469
Used	557	493
Increase	44	24

Table 22 – Effect of Chromium and Manganese Compounds in Lubricating Oil on Combustion-Chamber Deposits
(Controlled field tests; isooctane fuel)

% Additive in SAE 10 Solvent-Extracted Midcontinent Base Stock	Equilibrium Octane Requirement	Deposits		
		Weight, g	Volume, cc	Density, g/cc
None	90	27.0	22.5	1.20
0.25% Cr naphthenate plus 0.25% Mn naphthenate	90	21.1	13.2	1.60

others[6] indicate that compounds containing chromium and manganese are promising as combustion catalysts for use in mineral oils. However, in a controlled field test carried out with isooctane as the fuel, the same equilibrium requirements were obtained for the SAE 10 Midcontinent mineral oil A containing 0.25% chromium naphthenate and 0.25% manganese naphthenate as were obtained with the base stock alone. No preignition was observed with either oil. These data are summarized in Table 22.

It will be noted that the deposit weights and volumes are considerably lower for the cars operated on the lubricating oil containing the chromium and manganese salts. Despite this reduction in deposits, the equilibrium octane-number requirements are about the same. Why this reduction in deposits was not accompanied by lower requirements is not known. It does indicate, however, that the physical and chemical nature of the deposits, as well as the amount of deposits, may be an important factor in determining their effect on octane requirement increase.

A large number of other additive materials have been evaluated in the lubricant. To date none have been found that appear promising enough to warrant extensive study.

In summary, then, the data obtained with a noncontributing fuel indicate clearly the approach to making a noncontributing lubricant. This lubricant consists of using a low viscosity neutral distillate thickened with a suitable V.I. improver rather than with bright stock. To make a satisfactory finished lubricant, the formulation must also contain other additives, such as detergent-inhibitor, pour depressant, and so on, which are also selected to give as little increase as possible in octane requirement. Oils of this type can be made to have unusual temperature-viscosity relationships and to fit the SAE 5W-20 or 10W-30 viscosity classifications. It has been shown that, with a noncontributing fuel such as isooctane, such products have an octane requirement advantage of about 6-7 octane numbers over conventional SAE 10 motor oils. With a leaded commercial gasoline they also show a definite, although smaller, advantage. On the average, compared to a conventional SAE 10 mineral oil, this advantage will be as much as 4 octane numbers in equilibrium requirement. This will be true with respect to either the detonation or preignition requirement. The performance of such an improved product with commercial gasoline is discussed in the following section.

B. *Studies Using Commercial Gasolines* – 1. Effect of 5W-20 oil on detonation requirements: An

[6] "Combustion-Chamber Deposits as Related to Carbon-Forming Properties of Motor Oils," by J. D. Bartelson and E. C. Hughes. Presented before American Chemical Society, Atlantic City, Sept. 14-18, 1952.

Table 23 – Effect of Lubricants on Combustion-Chamber Deposits
(Controlled field test; commercial leaded fuel A)

	Equilibrium Octane Requirement[a]	Deposits	
		Weight, g	Volume, cc
Conventionally Refined SAE 30 Service Station Oil, MIL-O-2104	93	133.5	45.9
Solvent-Extracted Mineral Oil X, SAE 10	91	116.9	35.0
SAE 5W-20 Motor Oil[b]	87	78.0	27.6
Synthetic Oil Y	87	83.6	25.7

[a] Average initial requirement 78.
[b] Contains detergent-inhibitor additive R, V.I. improver M, and a pour depressant.

Table 24 – Effect of Bright Stock on Combustion-Chamber Deposits
(Controlled field tests; commercial leaded fuel)

% Bright Stock in SAE 5W-20 Motor Oil	Equilibrium Octane Requirement[a]
0	87
5	91

[a] Average initial requirement 78.

Table 25 – Effect of Lubricants on Octane Requirement
(Overhead-valve, v-type engines; leaded premium fuel; 6000-7000 miles)

	Employee Cars[a]		Controlled Field Test	
	Equilibrium Requirement	Advantage for 5W-20 Oil	Equilibrium Requirement	Advantage for 5W-20 Oil
SAE 5W-20 Oil	86	—	87	—
SAE 5W-20 Oil + 5% Bright Stock	90	4	91	4
Conventionally Refined Paraffinic SAE 10, MIL-O-2104 Oil	91	5	93[b]	6[b]

[a] 4-car averages.
[b] For the SAE 30 grade of the same brand oil.

Table 26 – Effect of Lubricant Composition on Preignition
(Controlled field tests)

	Commercial Leaded Fuel		Isooctane	
	Equilibrium Octane Requirement	% Observations When Preignition Exceeded Detonation	Equilibrium Octane Requirement	% Observations When Preignition Exceeded Detonation
SAE 5W-20 Motor Oil	87	17	83	None
SAE 5W-20 Motor Oil + 5% Bright Stock	91	30	—	—
Solvent-Extracted Mineral Oil A, SAE 10	91	70	90	None
Conventionally Refined SAE 30, MIL-O-2104	93	40	—	—

improved finished lubricant based on the principles established with a noncontributing fuel has been widely evaluated using leaded commercial fuels. This product, referred to as an SAE 5W-20 motor oil, shows a significant advantage over previously available more conventional oils with respect to combustion-chamber deposits and octane requirement increase.

In controlled field tests, the SAE 5W-20 motor oil was compared to: (1) a conventionally refined SAE 30 service station oil, made from a highly paraffinic crude and meeting the MIL-0-2104 requirements, (2) the straight SAE 10 mineral oil X used previously, and (3) the synthetic oil Y. The commercial high-quality premium gasoline A containing about 50% catalytically cracked stocks and about 2 cc tetraethyl lead per gallon was used as fuel. As illustrated in Table 23, the SAE 5W-20 motor oil had a considerable advantage in equilibrium octane requirement over the other mineral lubricants and was fully equivalent to the synthetic motor oil Y.

The weights and volumes of the deposits shown in Table 23 are in good agreement with the observed octane-number requirements.

The adverse effect shown previously for bright stock in the lubricant on octane requirement is also observed with leaded fuels, as shown in Table 24. With a 5W-20 motor oil containing no bright stock, the equilibrium requirement was 87, whereas the same oil containing 5% bright stock gave an equilibrium requirement of 91.

To confirm the results of these controlled field tests, several oils were evaluated under "normal" driving conditions in cars owned and operated by some of our employees. These cars, of the same make as used in the controlled field tests, were operated in the Elizabeth, New Jersey, area using a premium-grade, service station gasoline. The combustion chambers of these cars were cleaned at the start of the test and the crankcases were flushed out with the test oil. Four cars were operated on each oil. Octane requirement data were obtained on the cars monthly or every 500-600 miles. The cars were run to equilibrium octane-number requirement, which was reached in about 5000-9000 miles. The data from these experiments were in good agreement with the data from the controlled field tests, both with respect to the level of octane requirement observed and the advantage for the SAE 5W-20 oil over the other more conventional oils used. This is illustrated in Table 25.

It will be noted that, as in the case of the controlled field tests, the effect of bright stock is still easily recognized. All of these data show that the basic principles established with a noncontributing fuel are still valid when using a leaded commercial gasoline in average driving.

2. Effect of SAE 5W-20 oil on preignition: Because of the mild conditions used in the controlled field test, preignition occurred more frequently than would be normally observed. At the time the cars were rated, observations were made as to whether preignition did or did not occur. This was done on an aural basis. When preignition occurred it was the wild-ping type and required fuels 1 to 2 octane numbers higher than would satisfy detonation requirements to give "noise-free" operation. This, of course, means that for a lubricant or fuel to be really effective in reducing engine octane requirement, it must reduce or prevent preignition as well as detonation.

As shown in Table 26, preignition did not occur at any time when isooctane was used as the fuel. Since the SAE 5W-20 motor oil used contained detergent additives, this indicates that metal-containing detergent additives do not necessarily cause preignition. This may, of course, depend on the nature of the additive employed. When leaded fuels were used, preignition was apt to occur. However, it occurred less frequently with the SAE 5W-20 motor oil than with the other types of lubricating oils evaluated. With the 5W-20 oil used, the

preignition requirements exceeded detonation requirements in only 17% of all the octane requirement ratings made. Adding bright stock to the oil raised this value to 30% of the time. The solvent-extracted SAE 10 mineral oil X gave preignition in 70% of the octane requirement ratings made. With the conventionally refined SAE 30 paraffinic oil, preignition was limiting 40% of the time. However, because of the high octane requirement with this oil, this value may be more critical than a larger degree of preignition at lower levels of octane requirement.

The data in Table 26 show that the SAE 5W-20 oils can be made to have a definite advantage over previously available mineral-based compositions with respect to preignition as well as detonation. This aspect may be more important if preignition becomes a more critical field problem with higher compression ratio engines than it is now.

A number of products of the SAE 5W-20 type are now on the market. The first one was introduced about a year ago. In general, these products are getting excellent acceptance from the consumer and the automotive industry in general.

Conclusions

The following conclusions can be drawn from the data presented.

1. To evaluate the performance of a lubricant adequately with respect to its effect on car octane requirement, a clean-burning, noncontributing fuel must be used to eliminate complicating fuel effects. Isooctane is such a fuel.

Similarly, when evaluating fuels, lubricant effects must be eliminated by using noncontributing lubricants, such as a low viscosity mineral oil thickened with a noncontributing V.I. improver, or selected types of synthetic motor oil.

2. The most important factor in determining the extent to which a lubricant contributes to combustion-chamber deposits and octane requirements is its molecular weight or volatility. Bright stock is particularly harmful. Crude source (or variation in predominating hydrocarbon type) seems to have little effect.

3. Commercially practical mineral-base oils have been developed that show a significant advantage over previously known conventional lubricants in their ability to prevent combustion-chamber deposits and octane requirement increase when used with a conventional leaded fuel. It has been shown that SAE 5W-20 motor oils can be made that give a 4-5-octane-number advantage over conventional motor oils in preventing detonation. Such oils have also been shown to give less preignition than conventional motor oils.

4. While progress has been made in reducing octane requirement increase and combustion-chamber deposit formation by the development of improved lubricants, there is further room for improvement. The complete solution of the problem requires also an improved fuel. Both of these areas provide interesting possibilities for further research on this very important problem.

Acknowledgments

The results discussed in this paper represent the efforts of a number of other workers at the Esso Laboratories in addition to the authors. The authors take this means to express their appreciation and recognition of these contributions.

APPENDIX

A. Controlled Field Test Technique – The experiments reported in this paper were carried out for the most part in a test fleet consisting of sixteen 1951 cars equipped with an overhead-valve, V-type engine of 7.5/1 compression ratio, operating under controlled conditions. At least two cars were run in any given experiment on each fuel and lubricant combination. Three hundred pounds of ballast were added to the cars to simulate an average load in addition to the driver. The cars were operated in two caravans of eight cars each on a cycle in and around the city of San Antonio. The drivers were rotated among the cars to eliminate possible driver effects. The cycle consisted of 50% traffic driving in downtown San Antonio and 50% intercity-type driving on the outskirts of San Antonio, on a mileage basis. A top speed limit of 45 mph was imposed on the cars at all times. The average speed of the cars during the tests was 22 mph. The engines were equipped with full-flow oil filters, which were changed every 5000 miles. The oil was changed at 2000-mile intervals. The combustion chambers were cleaned at the start of every experiment. In addition, the oil pans were removed and any sludge or varnish deposits were cleaned out prior to installing the test lubricant.

The cars were rated at 800-1000-mile intervals using primary reference fuels by the standard Uniontown procedure. The occurrence of preignition, as well as detonation, was noted and recorded during these ratings. The experiments were continued for 2000-4000 miles after the cars had reached deposit equilibrium, as evidenced by a leveling off of the octane requirements. In general, equilibrium was reached at about the 3000-mile point, and the experiments were continued to 5000-7000 miles. The average of the last 3 to 5 ratings in any experiment is reported as the equilibrium octane requirement.

B. Field Test Reproducibility – The results obtained under the controlled field conditions used in this study are quite reproducible, both with respect to octane requirements observed and with regard to the amount of deposits formed in the combustion chambers. For example, a combination of the typical commercial gasoline (A) containing about 2 cc

Table 27 – Effect of Fuel and Lubricant Composition on Octane Requirement Increase
(Controlled field test results)

Experiment No.	Octane Requirements		Octane Requirement Increase	Combustion-Chamber Deposits per Engine	
	Initial	Equilibrium		Weight, g	Volume, cc
		Commercial Fuel A – Synthetic Lubricant Y			
1	76	86	10	54.6	17.0
2	80	88	8	61.8	19.2
3	75	86	11	54.1	16.5
6-Car Average	77	87	10	56.8	17.6
		Isooctane Fuel – Synthetic Lubricant Y			
4	80	77	−3	1.0	0.6
5	79	79	0	3.5	1.2
4-Car Average	79.5	78	−1.5	2.3	0.9

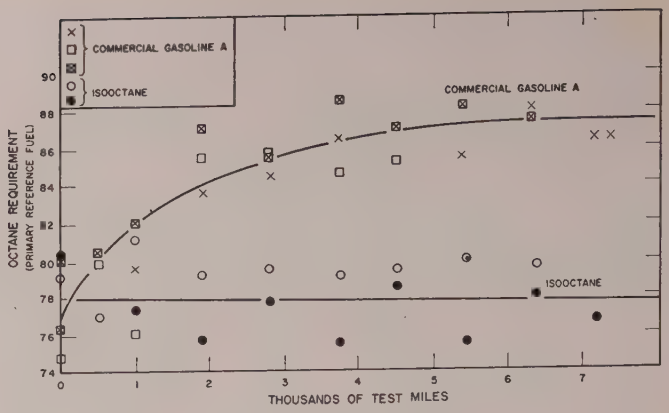


Fig. 8 – Increase in octane requirement with test-miles – overhead-valve, V-type engines; synthetic lubricant Y – each point is 2-car average

of tel per gal and the synthetic motor oil (Y) was evaluated on several different occasions, each time in two cars. Typical data are shown in Table 27 and Fig. 8. It can be seen that the cars came to equilibrium in the range of 86 to 88 octane requirement (2-car averages) and the deposit weights on a 2-car average basis varied from 54.1 g to 61.8 g per engine. The volume of the combustion-chamber deposits obtained in each case was almost the same, varying only from 16.5 to 19.2 cc per engine. Similarly, very good reproducibility was obtained when using isooctane and the synthetic lubricant Y. In this case, no requirement increase occurred and the equilibrium requirements were, therefore, about the same as the initial requirements. The combustion-chamber deposit weights were very low, being of the order of 2 g per engine. It appears, therefore, that this is a reproducible test, both with respect to octane requirement levels reached and with respect to the amount of deposits formed, irrespective of whether large or small octane requirement increases and deposit formation occurred.

Data for the individual cars in Table 27 are shown in Table 28. It will be noted that reasonably good reproducibility is obtained, even with individual cars. It is believed that the repeatability of any single determination using 2 cars is on the order of ± 2 octane numbers.

The volumes shown in Tables 27 and 28 were measured by a gas compression method that has been reported previously.[7] Briefly, the method is as follows: The pressure required to compress the air in a small glass cylinder by a fixed amount (volume) is measured, first with the cylinder empty, and second with the cylinder containing the deposits. The volume of the deposits is then readily calculated from the gas laws.

[7] CRC Symposium on Combustion-Chamber Deposits, Dearborn Inn, Mich., September, 1952.

Table 28 – Octane Requirements with Accumulated Miles
(Overhead-valve, V-type engines – synthetic motor oil Y)

Phase No.	Car No.	Octane Requirement at									Total Deposits per Engine	
		Start	500 Miles	900 Miles	1800 Miles	2700 Miles	3500 Miles	4400 Miles	5300 Miles	6200 Miles	7100 Miles	Weight, g	Volume, cc
						Commercial Leaded Fuel (A)							
I	A	76.0	—	79.0	82.5	82.0	85.5	85.5	85.5	90.0	82.5	49.9	17.7
I	B	76.5	—	80.0	85.5	87.0	87.5	85.5	85.5	86.0	90.0	59.3	16.3
		76.3		79.5	83.8	84.5	86.5	85.5	85.5	88.0	86.3	54.6	17.0
II	A	77.5	80.0	82.5	87.5	86.5	88.5	87.5	86.5	86.5	—	66.6	20.7
II	B	82.5	81.0	81.5	86.5	85.0	88.5	86.5	90.0	88.5	—	57.0	17.6
		80.0	80.5	82.0	87.0	85.7	88.5	87.0	88.2	87.5		61.8	19.2
III	C	76.0	80.0	77.5	88.5	89.0	86.0	86.0	—	—	—	51.3	16.7
III	D	73.5	80.0	74.5	82.5	82.5	83.5	85.0	—	—	—	57.0	16.3
		74.7	80.0	76.0	85.5	85.7	84.7	85.5				54.1	16.5
						Isooctane Fuel							
I	F	81.0	—	76.5	76.5	76.5	77.5	79.0	77.5	78.5	76.0	0.8	0.7[a]
I	G	79.5	—	78.0	75.0	79.0	73.5	78.0	73.5	77.5	77.5	1.3	0.5[a]
		80.2		77.3	75.7	77.7	75.5	78.5	75.5	78.0	76.7	1.0	0.6
II	H	80.0	77.5	79.0	78.5	80.0	78.5	77.5	79.0	77.0	—	3.1	0.9
II	I	78.5	76.5	83.5	80.0	79.0	80.0	80.0	81.0	82.5	—	4.0	1.5
		79.2	77.0	81.2	79.2	79.5	79.2	78.7	80.0	79.7		3.5	1.2

[a] Because of small amounts the volumes were estimated rather than measured.

Development of an Improved Automotive Diesel Combustion System

Bruno Loeffler, Mack Motor Truck Corp.

This paper was presented at the SAE National Diesel-Engine Meeting, Chicago, Nov. 4, 1953.

FUEL economy, although always the principal attraction in automotive applications of the diesel engine, has become increasingly important of late years. This is due to advances made in economy of gasoline engines, the increasing interest in LPG adaptations, and the general rise in all items of motor transport expense. At the same time, in this country at least, users are unwilling to sacrifice flexibility, adequate power output, reasonably light weight and compactness, smoothness, smokelessness, and low maintenance expense for the sake of low fuel consumption.

For these reasons Mack, for several years, has been developing a combustion system which, while retaining these latter virtues, achieves definite improvement in fuel economy. After prolonged laboratory experiments and tests, a considerable number of engines embodying this new direct-injection combustion system were subjected to extensive field testing, beginning in the winter of 1952 and continuing into the summer of 1953. Then the new engine was placed into full production, superseding an engine of the indirect-injection type which has been successfullly produced for 12 years.

Naturally, the first step was to explore thoroughly the present developments both here and abroad. To this end automotive technical literature was painstakingly reviewed, and practice in England and on the Continent was investigated first hand. The author spent three months in Europe to interview leading engineers responsible for diesel design. The discussions and observations were enlightening both as to the real progress which is being made overseas and also in revealing some of the fundamental differences of approach and in the problems facing designers overseas as compared with the situation here in America. Particularly helpful were the discussions with Mr. C. B. Dicksee of A.E.C. of England and Mr. F. Hedlund of Scania-Vabis, Sweden.

Our present combustion chamber of the single-lobe energy-cell type was developed some seven years ago. Before that time, we employed the double-lobe design. Many thousands of engines have been built using both combustion designs.

With the single-lobe chamber design, we have achieved the desirable features required for an automotive-type diesel engine. These features consist of:

1. Flexibility and smooth operation.
2. Durability and economy.

By flexibility we mean the possibility of having the operational range from 1400 to 2000 rpm with-

INCREASING fuel prices and fuel taxes have forced engine builders to find ways to better engine specific fuel consumption in order to help operators survive the onslaught of increased operating costs. This paper explains how an improved combustion system has yielded up to 15% improvement in fuel consumption.

This system was arrived at by analysis of those previous combustion systems which held most promise. The resulting direct-injection open combustion chamber is described here.

The Author

BRUNO LOEFFLER (M 42) is executive engineer, Engine Division, Mack Mfg. Corp. He graduated from the Technical College at Winterthur, Switzerland in 1917 and came to America in 1920. Mr. Loeffler, who holds more than 30 patents applying to internal-combustion engines, started with Mack in 1922 as a draftsman and became chief design engineer in 1942.

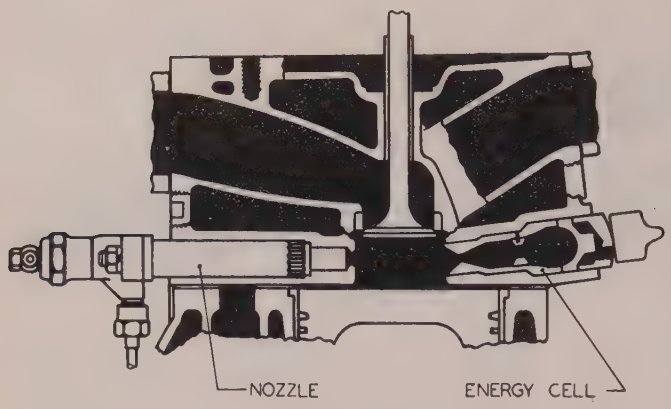


Fig. 1 – Section through energy-cell cylinder head

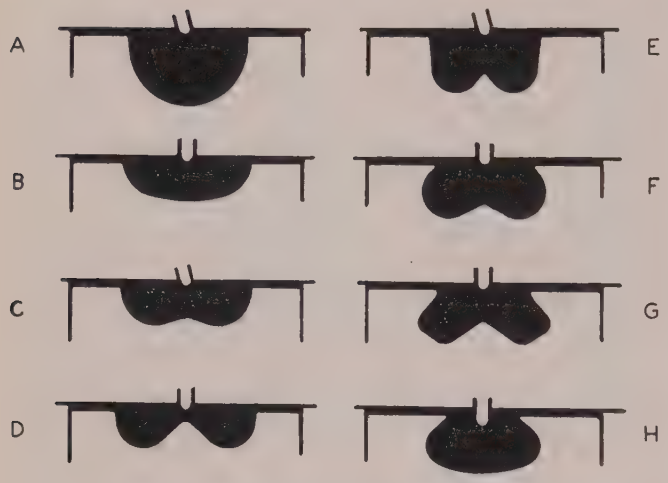


Fig. 2 – Forms of British combustion chambers used in automotive diesel engines

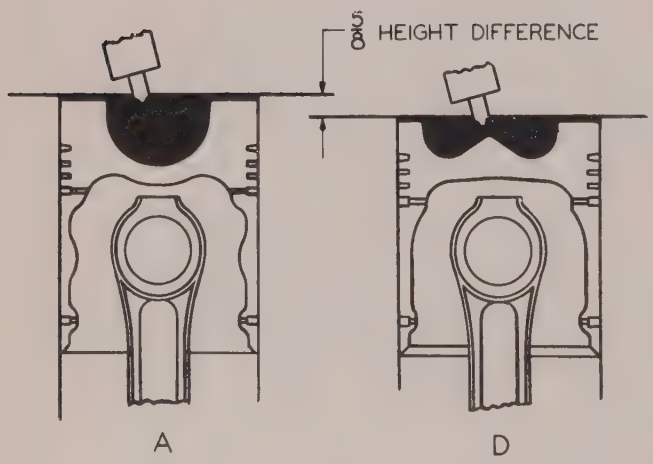


Fig. 3 – Chamber design affects weight and height of engine

out excessive smoke and engine knock which destroys ring setup.

Flexibility and Smooth Operation – Smooth combustion is achieved by means of properly proportioned energy-cell volumes and orifice size. The importance of the angle of inclination as well as the location within the combustion chamber should not be forgotten.

Durability – By means of modifications in energy-cell proportions, maximum pressures can be reasonably controlled. To improve low-speed economy, knocking at lower speeds, and as a starting aid, an automatic advance device is incorporated ahead of the fuel injection pump. Because of it, the maximum pressures at the lower speed range are greatly reduced, thus reducing the breaking of piston rings and knocking out of bearings. These items improve durability.

Economy – The economy, that is specific fuel consumption, is in line with other competitive engines of equal size and at times surpasses them.

Starting

Starting response in cold weather with this type of combustion chamber was somewhat inferior to that of open-chamber types, so that special means had to be provided, such as manifold heaters or ether capsules, in order to obtain starts at low temperatures.

These, then, were the characteristics of the energy cell of our existing combustion system. (See Fig. 1.) We realized that any improvements we wanted to achieve would have to come from modifications thereof.

Our main object was to improve fuel economy as well as the starting conditions without losing the desirable features we now have such as horsepower, flexibility, durability, and reduction of combustion noise and smoke.

The reason for concentrating on fuel economy was the increase in fuel cost per gallon. In addition to this price increase, taxes on diesel fuel have been imposed or increased. Therefore, fuel economy becomes increasingly important. However, when we started this program in 1949, we did not expect that in 1953 fuel prices and taxation would be raised to such an extent that would seriously affect the further usage of diesel engines.

We had to avoid expenditure of energy in the creation of the air movement needed to obtain complete combustion.

Direct injection into an open chamber located in the piston crown seemed to offer the greatest possibilities.

The art of combustion-system design for automotive diesels is quite well developed in England. There, of course, fuel economy is of prime importance because of exorbitant taxation. It was just good business sense for us to look into the prior art of their combustion systems in order to find the best type for our application.

In examining technical literature of the past few

years, we found a great variety of combustion-chamber designs. Most of these were located in the piston head. Some representative forms are given in Fig. 2.

All of these chambers are being produced and give specific fuel consumption well below 0.4 lb per bhp-hr at their maximum rated power and speed. Best economy lies near the 0.350 lb per bhp-hr mark. This is truly remarkable economy in comparison with that of American diesels.

Upon completion of our analysis of the eight different shapes of chambers shown, we arrived at the following conclusion.

Chambers F, G, H were eliminated because their restriction of the mouth of the chamber hinders proper scavenging of the exhaust gases. These chambers resemble more the swirl-type variety. After all, direct injection is interpreted as having a dished piston with no constriction in its communication with the clearance space.

Moreover, machining of these chambers to correct volumes, while not insurmountable, becomes rather expensive.

Designs A and E fall into the open-chamber class. Both, however, require long pistons to accommodate the deeper recess. The piston becomes top heavy, the cylinder height is increased, and the reciprocating masses are also slightly increased. Because there are no restrictions, scavenging is improved over the previous designs. Aside from the above points, they should perform well.

Designs B and C are of the shallow type, and scavenging is very much better because of the open form of chamber. They should show good volumetric efficiency.

However, merely to achieve good volumetric efficiency is not enough. Unless every bit of the air is put to good use, the desired power output will fail to materialize. In these designs, the air directly below the nozzle is relatively quiescent, and the spray does not reach it. This is an obvious detriment.

In design D this deficiency is eliminated. This chamber, which has a rather large diameter and shallow depth, is often referred to as the "Mexican Hat" type.

The center section, just below the nozzle, is formed as a raised cone. This places the maximum of air in the periphery where the fuel sprays can reach it.

The shallow design also has the advantage as to engine height and weight. This is shown in Fig. 3.

As laid out for our case, the cylinder and piston weight would have been raised by 10 lb, the height by $\frac{5}{8}$ in.

Quiescent Chamber

A quiescent chamber shown in Fig. 4 can be found in larger bore engines. There is no controlled air movement outside of the movement established by the inrushing air through the inlet port. Because

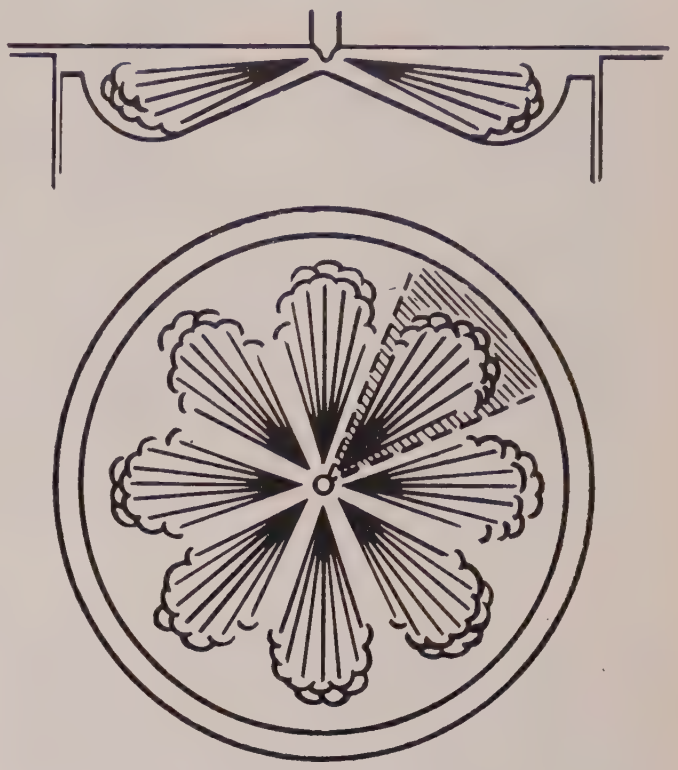


Fig. 4 – Quiescent chamber requires more and smaller spray holes to reach all air particles

the fuel must reach the quiescent air, a large number of spray holes are needed. The quantity depends upon the segment of air. If the segment is too large, not all surrounding air can be reached by the fuel. This means the available air is not fully utilized.

When many holes are needed, they must become small in order to keep the velocity through the spray holes high. While there is no hardship in drilling small holes, they add to the cost of the equipment. Naturally, such small holes are apt to clog more easily than those twice their size.

The air can be used more fully by passing it in review of the fuel spray.

Induced Swirl

When high output at high speeds is required, air movement is essential. This air movement must be of an orderly nature and of a velocity that assists the injected fuel to be burned completely.

This air movement, or swirl, can be induced by a masked inlet valve or by an inlet passage so disposed that it imparts direction of airflow tangential to the cylinder and slightly downward, together with locally restricted port opening clearance (Fig. 5). Both arrangements are used with success.

We have tried both types and like the venturi

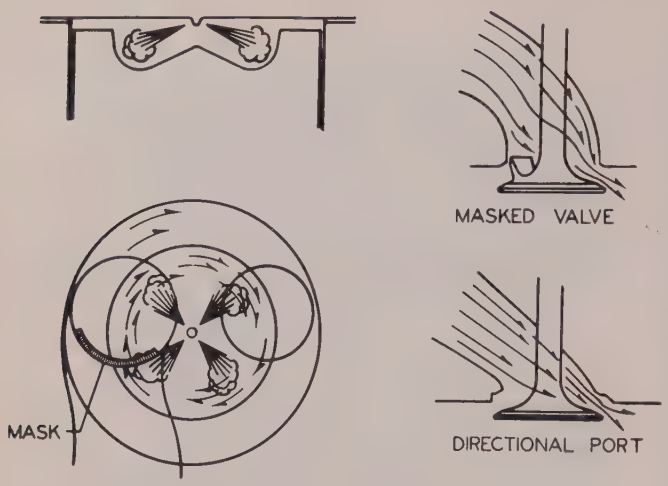


Fig. 5 – Intake induced air swirl and methods of producing same

port better. It allows the inlet valve to turn by means of a positive-type rotator which will keep the valve seat clean and reduces valve-stem scuff. This cannot be accomplished with a positioned inlet valve. There is very little argument as to which gives a higher volumetric efficiency. Both types will give a slight drop in air delivery.

Mr. Ricardo, the eminent British engineer, has made an analysis as to the swirl velocity which will give best power and economy. For his test, he used a sleeve-valve engine and varied the air entrance angle.

He built an air-speed indicator inside this chamber and, by motoring, established various velocities versus entrance angle. Subsequent engine tests gave him the optimum velocity. The best air rpm/crank rpm is between nine and ten. That is, the air inside the chamber, before combustion, revolves at nine to ten times the engine rpm. Since the trials were conducted in a sleeve-valve design, with the chamber located in the head and consequently stationary, the measurements could be taken in a simple manner.

In our case, where the chamber is in the piston head and, therefore, constantly in motion, such a device was not practical. Accordingly, we had to do some deducing from the trials with the masked valve as to the possible air speed.

We relied rather upon the cut-and-try method to determine the optimum. The masked-valve data helped us a great deal.

Squish

Squish is an air movement that takes place within the combustion chamber just after the fuel is injected. (See Fig. 6.) The air trapped in the quench area is squeezed out of its space at an accelerating rate and enters into the combustion chamber. The body of air being thus displaced into the chamber and into the already swirling air increases the air velocity. This assists the swirl to devour the fuel droplets which have presently been injected into this vortex of hot air.

The ratio of chamber diameter to piston diameter determines the squish velocity as shown in Fig. 7. Also affecting this velocity is the distance remaining between head deck and piston when the latter is in its uppermost position. The effect on

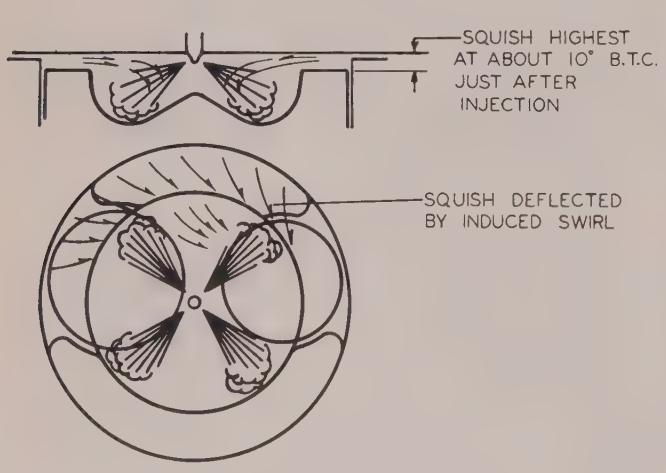


Fig. 6 – Piston induced squish

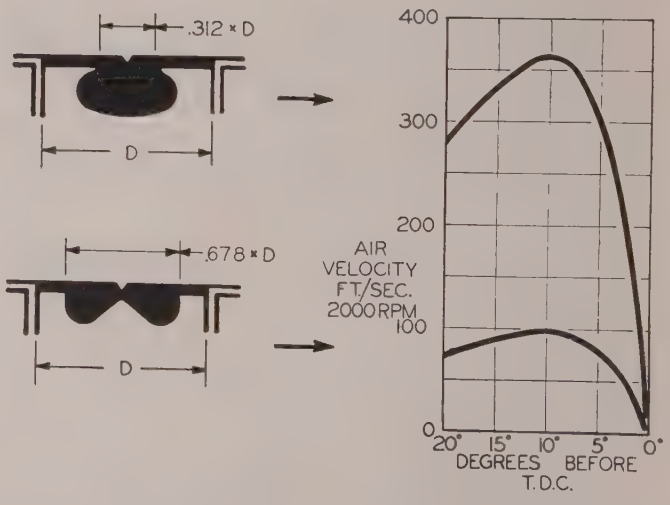


Fig. 7 – Chamber-opening diameter as it affects squish velocity

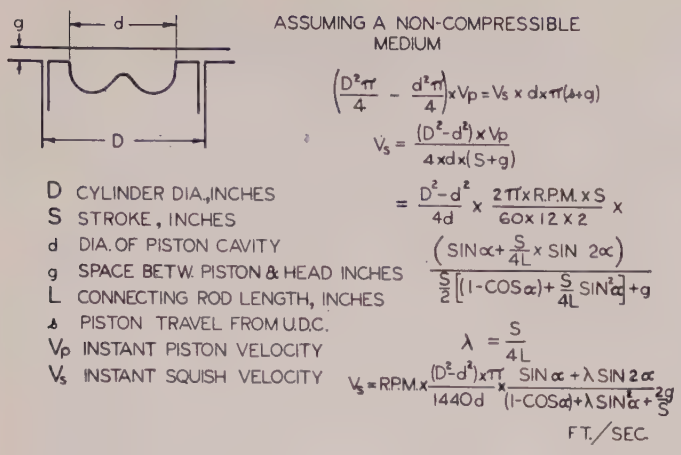


Fig. 8 – Abbreviated squish-velocity formula

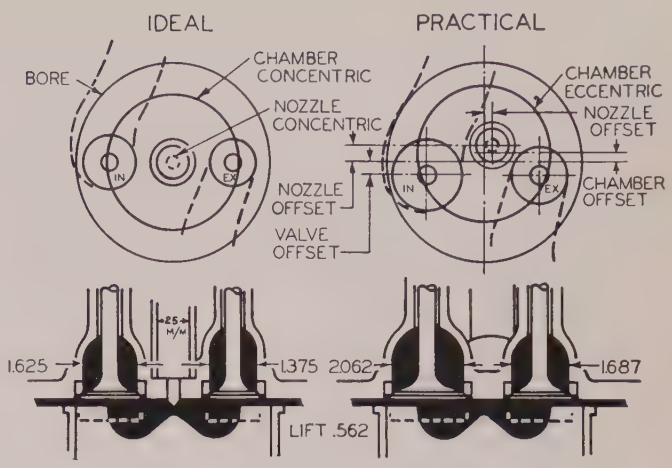


Fig. 9 – Ideal and practical valve and nozzle arrangements with their effect upon valve size

velocity with reduced chamber diameter can also be seen. The maximum point of velocity occurs at about 9 to 11 deg btdc.

We have a simple formula developed in order to arrive at these velocities. It assumes, for convenience, that the air is incompressible. Of course, it is not. For our comparison, it was sufficient to show certain effects by changes in the chamber proportions. Another item affects squish velocity as well. It is the relation of stroke to connecting-rod length.

This factor exerts such a small influence that it can safely be ignored. However, the height of the quench space has a very important effect upon squish velocity. Obviously, the closer the piston rim approaches the cylinder-head deck the greater the squish velocity. Naturally there were mechanical limitations, such as head gasket thickness, piston expansion and rocking, connecting-rod and main bearing clearance, expansion differential between the block and the piston and connecting rod, and connecting-rod stretching at overspeed.

Fig. 8 gives the developed formula we have used for comparison purposes.

Designing the Head and Chamber

With the analysis completed, we started layout work of the head and chamber. Our aim was to make a head interchangeable with the present energy-cell type. For our combustion chamber, we selected, as previously mentioned, design D in Fig. 2.

The ideal head carries the nozzle in the middle and has the valves arranged as shown in the left view of Fig. 9.

This looks good until one tries to draw in the valves and inserts.

As can be noted from the figure, the inlet passage had a diameter of only 1.625 in. We needed at least 2 in. in order to have airflow equal to that with the energy-cell type.

With proper valve sizes, it was not practical to have the valves in line with the nozzle and on the engine centerline and still have the valves opening directly into the bore and of sufficient size. Fig. 9 shows the crowded condition in the design shown at the left, resulting in the restricted size of the intake port.

We did not want recesses in the cylinder liner. In fact, we could not because of the dry liner design in this engine. From the experience we have gained in our experimentation, it would have been a mis-

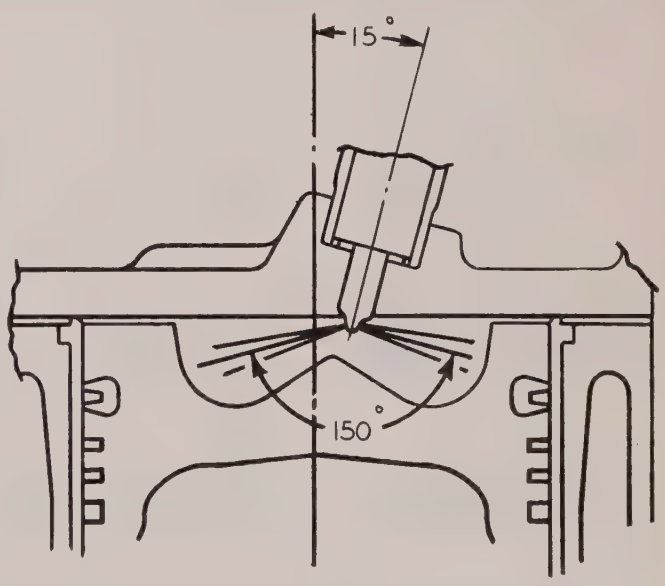


Fig. 10 – Nozzle and combustion-chamber relationship

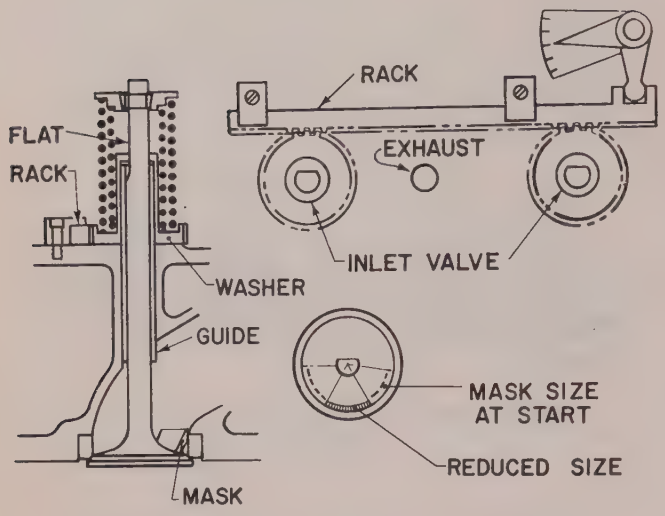


Fig. 11 – Schematic arrangement for mask positioning

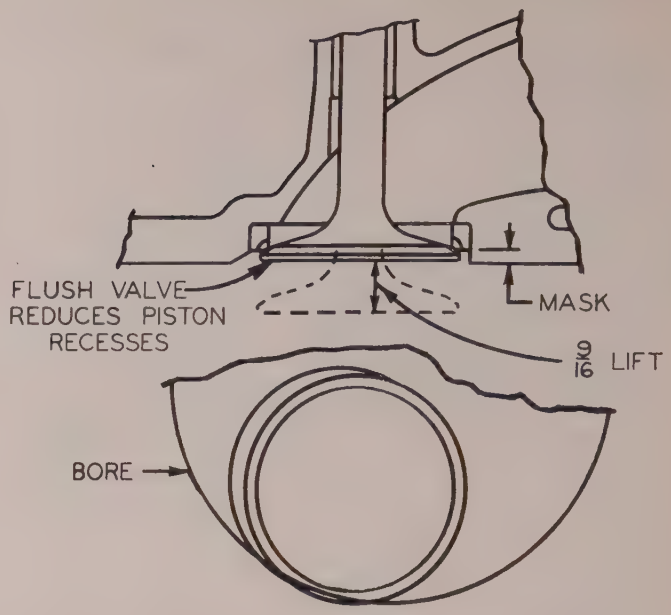


Fig. 12 – Intake port showing recessed insert and mask effect

take. It would have affected our fuel economy adversely.

To overcome this, the valves were moved to one side from the longitudinal centerline and the nozzle to the opposite side. To obtain good nozzle cooling, the nozzle was inclined to the maximum practicable.

Naturally with the nozzle tip so far off the center of the engine, we could not leave the combustion chamber in the center of the bore. We moved it close to the center of the nozzle, shown in Fig. 10. This offset the piston in relation to the nozzle location and resulted in more air being available on the left side of the nozzle tip. It just so happens that the two spray holes on the side pointing toward the more remote side of the chamber deliver about 2% more fuel than the other two. This may account for the lack of the smoke which is normally associated with off-center nozzle position.

In our early experimental work, we used a masked inlet valve to establish the best swirl velocity.

In Fig. 11, the size of the mask was purposely selected too large and was gradually cut down to a trifle.

In order to move the mask into various positions, all six inlet valves were hooked up by a rack and

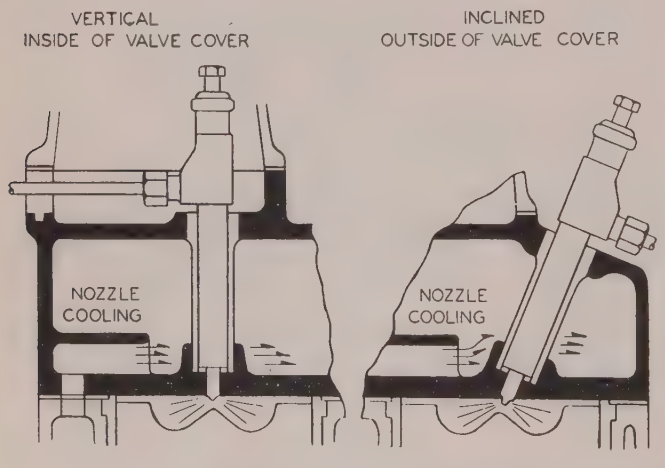


Fig. 13 – Vertical and inclined nozzle arrangement with method of cooling

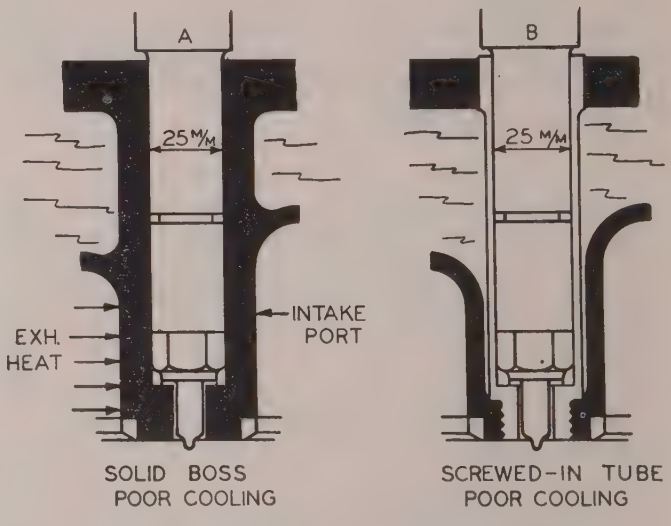


Fig. 14 – Various types of nozzle-holder tubes

pinion. This helped to reduce the down time for the various positions.

The idea is not novel. It was incorporated in the Saurer supercharged engine many years ago. In fact, it was used to intensify the swirl velocity in relation to manifold pressure on a turboblown engine. Apparently, with light loads, more swirl was necessary; while with heavy loads and high speed, the blower furnished sufficient pressure to augment the swirl.

When we modified the port, we removed the mask from the valve and incorporated a local restriction into the valve port, as shown in Fig. 12. The insert is recessed into the head deck. The resulting recess is cut away in the direction of the entering air. The opposite half of the recess is used to deflect the air. This gave us another advantage. It made the valve heads flush with the deck. The advantage is that the piston recesses disappear or become very small. Since the port entrance is flatter, it was later found necessary to increase valve lift to 9/16 in.; it was previously ½ in.

Nozzle Installation

As mentioned before, we decided to have the nozzle inclined. There are arguments for and against it. Fig. 13 gives us a comparison of the two arrangements.

Let's look at some of the arguments that can be advanced:

In favor of the inclined nozzle is the fact that nozzles can be removed for cleaning without disturbing the cylinder-head covers. This is not only a convenience on engines arranged vertically, but it also improves conditions on engines laid horizontally. Removing the head covers lets oil run over the engine, and the whole thing becomes messy. The least you disturb the head covers, the less dirt enters into the head mechanism.

Another advantage is the detection of joint leakage which might occur. There is no nasty packing for the fuel line to pass through the head wall. Leak-off lines are easily accessible and can be properly checked for leakage.

Of course, with the nozzle inserted from the outside of the engine, it is possible for a careless mechanic to allow dirt to enter the combustion chamber upon removal and replacement. This, however, is as true of the energy-cell type, which in the 15 years that we built this engine never proved serious.

What seemed at first to be a disadvantage of the inclined nozzle, namely, that the spray from the lower holes was some 2% greater in volume than from the upper ones, actually turned out to be an advantage in connection with the offsetting of the combustion chamber. The only penalty was the necessity for dowelling the nozzle holder to insure correct positioning of the spray holes.

This was a small price to pay for escape from the many drawbacks of the vertical nozzle. One of these is that with the nozzle under cover, possible joint leakage cannot readily be detected. The packing between the tube and head, mostly accomplished by neoprene blocks, is not durable.

Leak-off connections must also pass through head, again through packing. When fuel-packing leakage occurs under head cover, it cannot be easily detected.

Cooling of the nozzle tip is important. A copious flow of water directed against the tube, as shown, is necessary. This is not only necessary to the open chamber, but was an important item on the energy-cell type of head.

For the nozzle boss or tube, various designs have been evolved.

In Fig. 14, view A is probably the simplest arrangement. However, it has the disadvantage that the nozzle tip is buried in a lot of metal, and no water can take the heat away.

In fact, on the exhaust side, heat is continuously poured into the boss. It stands to reason that with heat on the exhaust side and cool air on the intake side, a warpage will take place. This is reflected upon the exhaust-valve insert seat being out of square. Nozzle cooling is thereby neglected. The nozzle performance may be adversely affected.

It is also very likely that heat from the exhaust port passes around the boss and heats up the incoming charge. This reduces the volumetric efficiency.

A somewhat improved design is represented in B. Here, the nozzle is in a tube which is screwed into the bottom deck. There is space between inlet and exhaust port for cooling. The bottom thread, however, with its sharp corners might easily crack this section. Heat transfer through a thread is at best only about 50% of a pressed-in tube.

In Fig. 15, view C shows a tube screwed into the

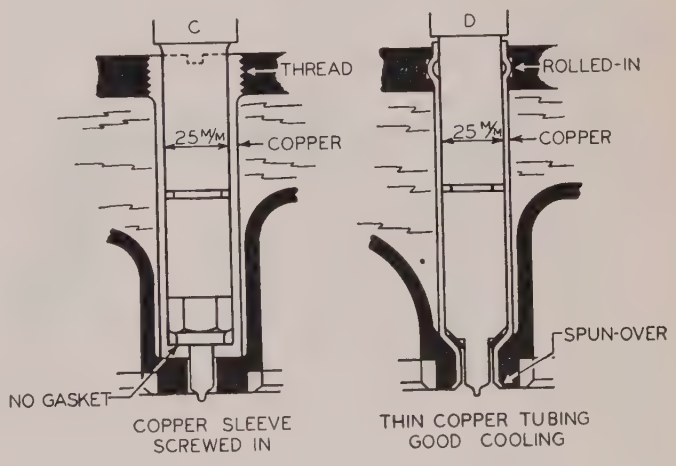


Fig. 15 – Various types of nozzle holders

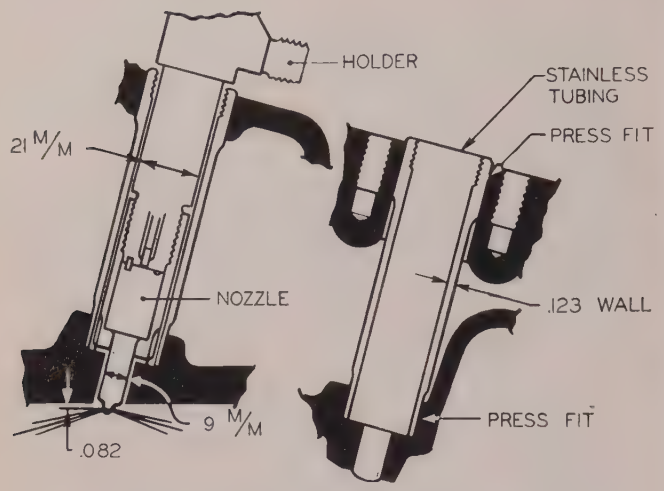


Fig. 16 – Finalized design of nozzle installation

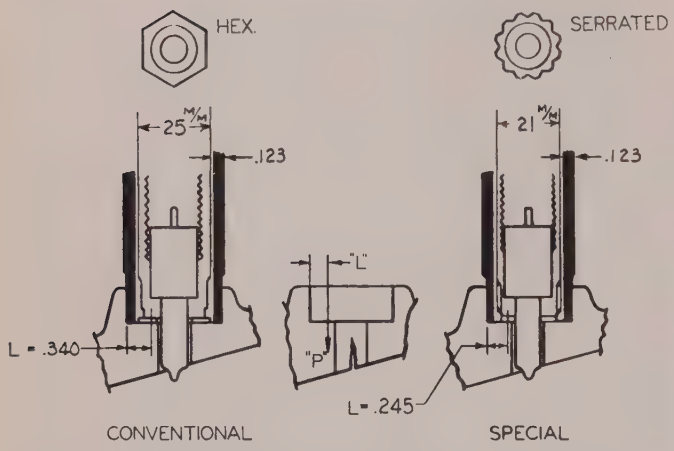


Fig. 17 – Nozzle nut design effect on reduction of stress at nozzle seat corner

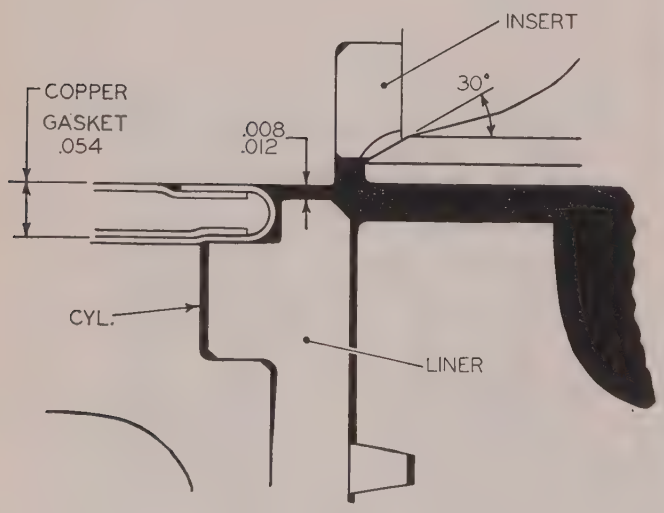


Fig. 18 – Means of protecting cylinder-head gasket from flame impingement

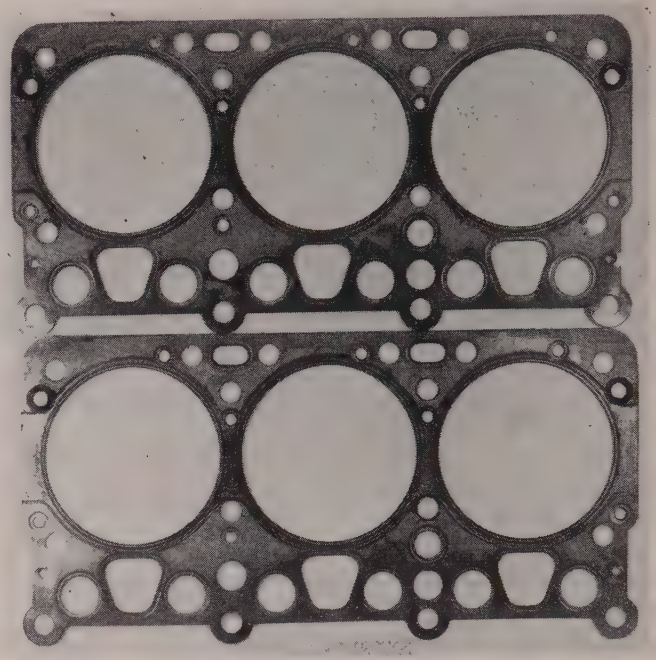


Fig. 19 – Appearance of gasket after severe cycle test

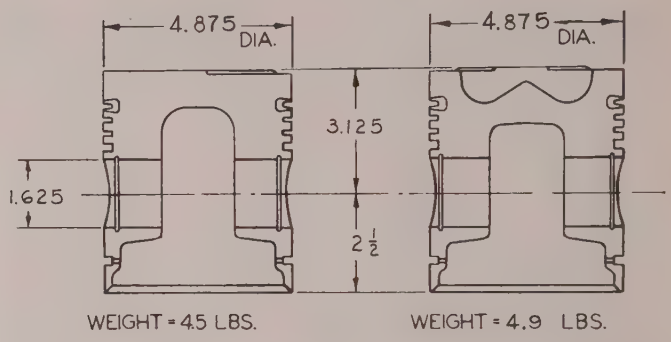


Fig. 20 – Piston design for energy-cell (left) and open-chamber (right) engines

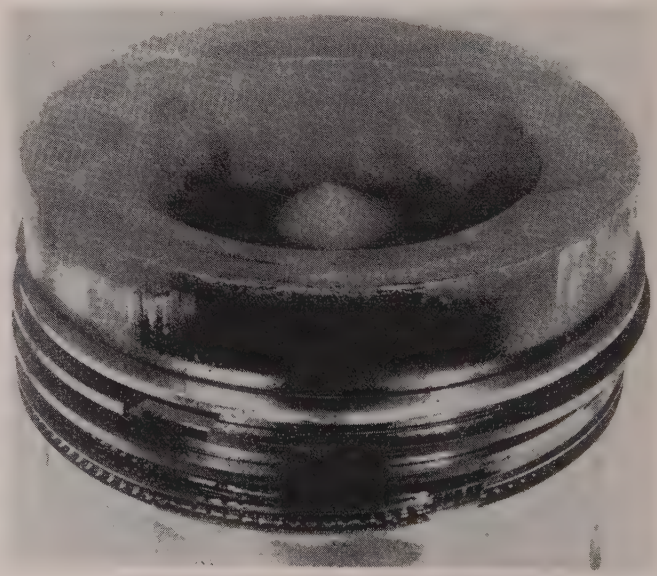


Fig. 21 – Typical piston condition after 140,000 miles of tractor operation

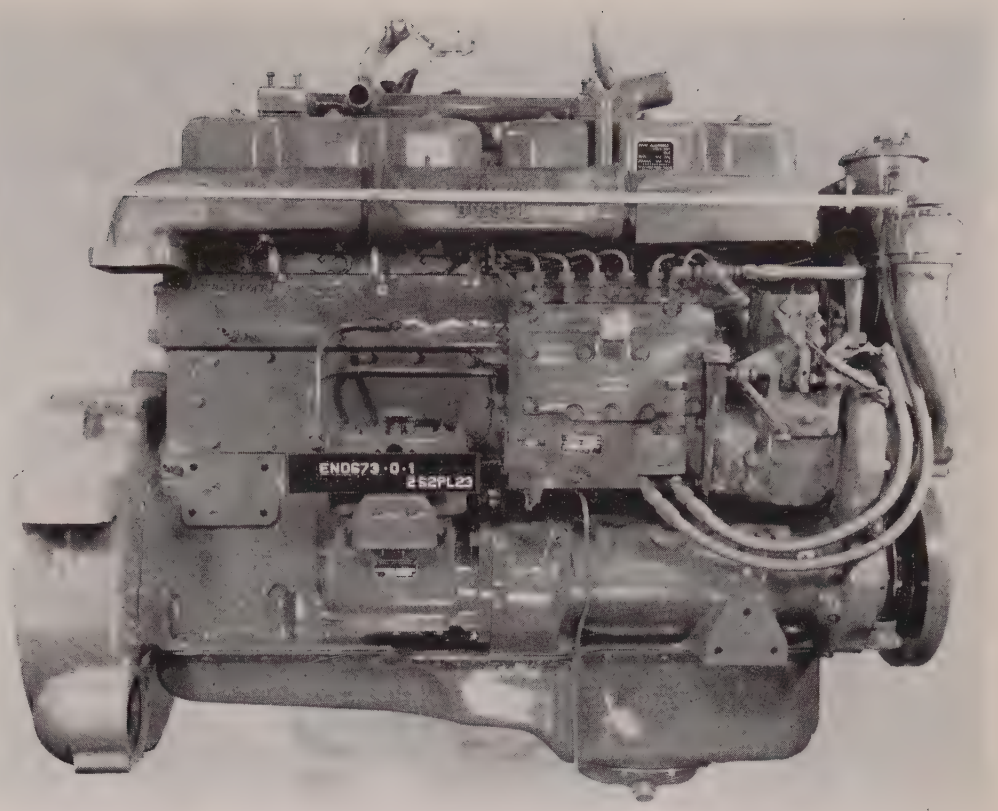


Fig. 22 – Mack END673 right side showing injection pump and governor

head. The threaded section is on top instead of the bottom.

The tube is normally made of copper. Expansion of copper in a heated head, however, will eventually bow the bottom or top deck. Since the sleeve is copper, no gasket is used to seal the nozzle. From our experience, it won't be long before the seat at the bottom of the tube will be deformed so that it must be replaced frequently.

Cooling, of course, is good. The copper will take a great deal of heat away from the nozzle seat.

The other extreme is design D where a thin tube of copper is used and spun over at the open end. The upper end is rolled into the recess cutout in the upper deck. Here a conical gasket is used. Again, this gasket leaves an imprint in the tube. After frequent removal of the nozzle, a ridge will form, and the tube must be replaced. The spun-over end is heated by the combustion temperature and elongates.

The amount of metal left between tubing and valve seat is very small. Every time a tube is inserted, valve seats must be reconditioned. Good nozzle cooling is obtained.

We have had some experimental heads equipped with such a tube but subsequently have gone to a different design, shown in Fig. 16.

Here a stainless-steel tube is pressed into a recess in top and bottom decks. Cooling is good. It offers solid metal for the gasket to rest on and, because of the tube material used, does not corrode or expand excessively. A thread is added to the top for removal with a puller.

With the conventional tube design, there is danger that the point of gasket contact (Fig. 17, left) is too far away from the bore. The force when fastening the holder may be excessive. This, coupled with long leverage, *L*, can split the head as shown on the center view.

To overcome or reduce this, the nozzle holder was made smaller, 21 mm instead of 25 mm, and a special serrated nut was used instead of the standard hexagon type, as in the right view. With these two features, the lever arm is reduced from 0.340 to 0.245, a sizable amount, and we do not expect any cracks at this place.

Gasket

We had considerable trouble at first with the gasket sealing. It might have been because we used steel selvage. The increased combustion pressures may have contributed. This pressure went up from 950 psi to over 1200 psi. After several failures of the selvage due to burning, we added a shoulder on our cylinder liner as shown in Fig. 18. It thus shields the gasket edge from the flame. This, together with a copper selvage, seems to have eliminated the trouble. Note that the gap between liner and head is from 0.008 to 0.012. This is a small space exposed to combustion flame. The reduction in volume also results in improved fuel economy. I want to stress the importance of the elimination

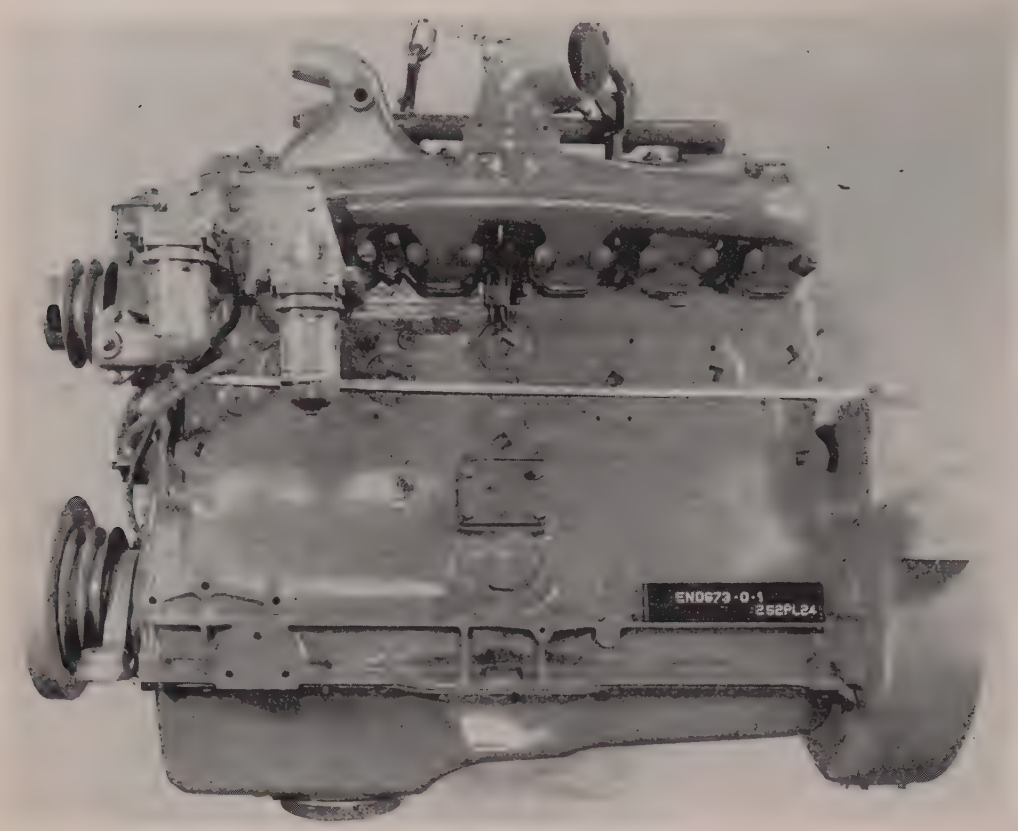


Fig. 23 – Mack END673 left side with primary fuel and governor oil filter

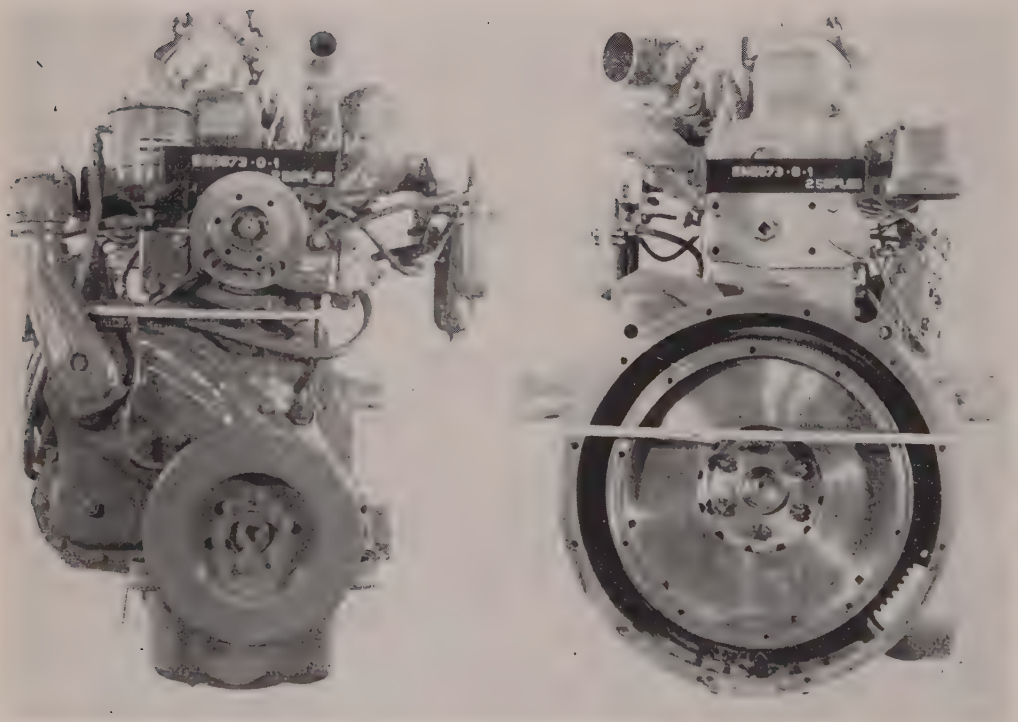


Fig. 24 – Mack END673 front (left) and rear views

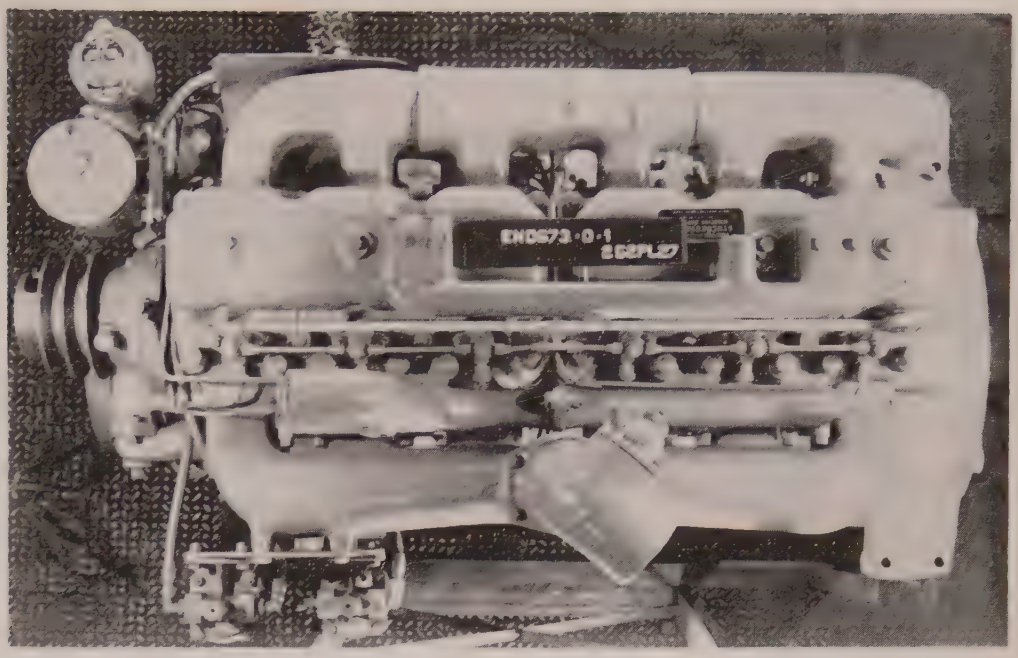


Fig. 25 – Mack END673 top view showing nozzle location

of all dead space such as the above. Pockets required in the piston crown are in the same class; when too deep, they affect economy. They reduce the effective quench area and, correspondingly, the squish velocity. Fig. 19 shows a photograph of two cylinder-head gaskets removed from a test engine after 92 hr of severe cycle testing.

Piston

Fig. 20 shows the comparison between the open-chamber and the energy-cell type piston.

The open-chamber type has the typical "Mexican Hat" shape, while the energy-cell type is flat. Both have the cast-in Ni-Resist insert and a keystone top ring; all remaining rings are identical.

Both pistons are of pretty rugged design to resist deflection.

We have found that ring sticking is greatly reduced by the keystone ring design. One would think that, because of the lowered ring belt temperature, ring sticking would not occur. We have tried straight rings and found the keystone rings a great improvement. Add to this the cast-in insert, which resists ring-groove pounding, the setup is quite good and resulted in longer ring life and improved oil consumption. In fact, after 140,000 miles of gruelling tractor operation, we still ran 300 mpq, and the top bore wear was just over 0.003 in.

Fig. 21 shows a photograph of one piston taken out of the same engine.

The Engine

The engine on which these tests were conducted is shown in Fig. 22. This is the right view and shows the model BB fuel injection pump and the hydraulic governor, below which is the housing for the automatic advance mechanism. The entire mechanism is enclosed and made tamper-proof by means of seals.

Fig. 23, the left side, shows clearly the inlet manifold and two filters, one of which is the primary filter, and the other is the governor lube oil filter.

The primary filter formerly was behind the cab, but winter operation necessitated moving it into a warmer climate. This stopped paraffin formation.

Fig. 24 shows front and rear views. The filter shown high up on the left side is the final-stage fuel filter in a very accessible position.

Fig. 25, the plan view, clearly shows the nozzle positions. One can see the manifold heater cup fastened to the elbow. On later series, we installed an ether starter unit which we have now standardized.

While the manifold heater works down to about

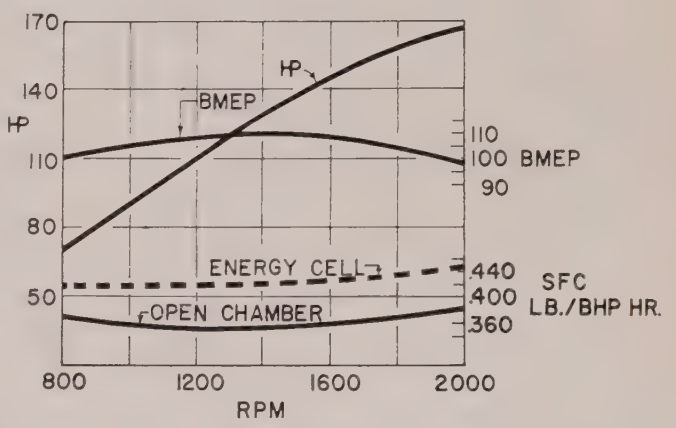


Fig. 26 – Power bmep, and fuel consumption for different combustion systems

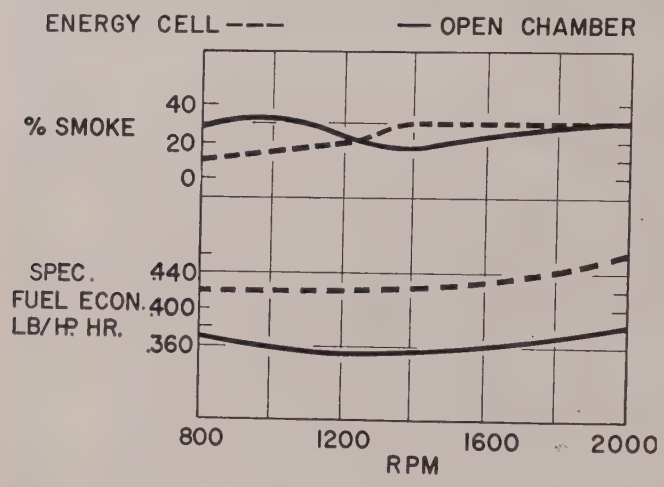


Fig. 27 – Economy versus smoke for energy cell and open chamber

Fig. 28 – Advertisement stressing smokeless operation

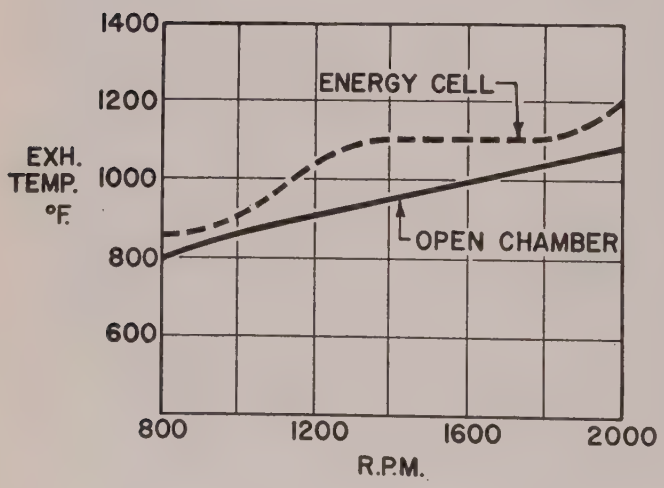


Fig. 29 – Exhaust-temperature comparison of two combustion-chamber designs

25 deg, we know that the operator needs a unit that can start his engine at a still lower temperature. With ether capsules, the engine can be started at very much lower temperatures as long as the battery is kept up. For continuous temperatures below zero, it might be necessary to have a good blowtorch handy.

Performance

Having the same bore and stroke, 4⅞ by 6, giving 672 cu in. piston displacement, the new engine also gives the identical horsepower and torque at the same speeds as the superseded engine, as shown in the curve of Fig. 26. This is 165 hp at 2000 rpm and 480 lb/ft at 1400 rpm but with improved economy. However, the improved heat conditions have permitted an increase in governed speed to 2100 rpm, at which the output is 170 hp, and the torque peak has been moved to 1200 rpm for the same 480 lb/ft.

The upper fuel consumption curve is that of the energy-cell-type combustion chamber, while the lower curve is that of the open-chamber design.

It does not need much imagination to see that a fuel economy, as shown, will give very much improved miles per gallon. The improvement over most of the usable speed range is better than 15%.

Only one power and brake mean effective pressure curve is shown. This is because the two power curves of both combustion chambers are so close together that the difference is not noticeable on this graph.

We established moderate power setting to minimize smoke that is spewed out by the mufflers.

In order to check each major move on the test engine as to good or bad combustion, we have built into each of our diesel engine laboratory dynamometer setups a CRC smokemeter.

Fig. 27 shows the smoke for the energy-cell and the open-chamber combustion systems. It should be understood that these observations are at full load and at all speeds. At reduced load, combustion is virtually smokeless.

An additional reason for stressing smokeless operation is the general public's indignation over the smoke nuisance on the highways. Our Sales Department also insists upon engine conditions that are not objectionable to the public. In fact, the Advertising Department has one picture that speaks for itself. This is shown in Fig. 28.

Naturally, smokeless operation means that the exhaust temperature is moderate. Fig. 29 shows the exhaust temperature under identical power. A lower exhaust temperature means more complete combustion. It signifies that no unburnt fuel remains within the combustion chamber or finds its way to the first piston ring.

Another way to analyze the fuel economy is by means of Fig. 30. Here the rpm is held constant and the fuel is gradually reduced. The top curve represents fuel economy versus brake mean effective pressure at 2000 rpm for the energy-cell com-

bustion system. A similar curve is just below for 1400 rpm for the same combustion system.

The third and fourth curves show the fuel economy obtained under identical conditions for the open-chamber system. Arrows point to the highest point of economy at each of the two speeds taken. Study of this graph reveals an unexpected gain which the open-combustion-chamber design has conferred. The points of maximum economy are reached at very close to maximum power output; whereas with the energy-cell type, maximum output is incompatible with utmost fuel economy. The advantages to the operator are obvious.

Meeting the exacting operating demands which prevail in this country necessitates keeping engine speeds between 1600 and 2000 rpm and at full power. This is why it is of such great advantage to have maximum economy within the prevailing operating range. The maximum economy is at 1400 rpm, from which point it falls off gradually. As a consequence, it will seldom be realized.

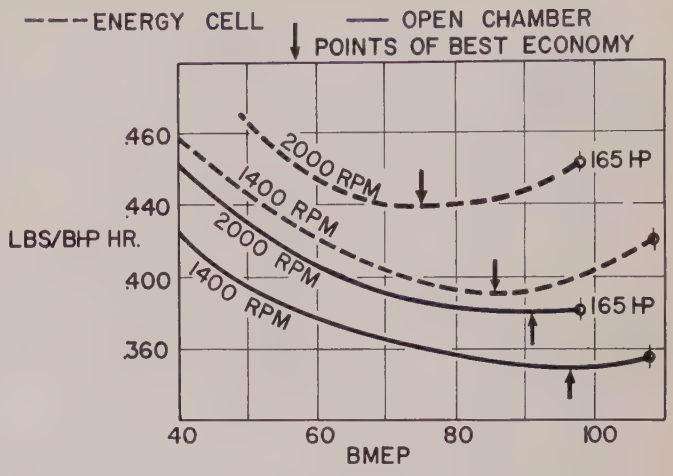


Fig. 30 – Comparative fuel economy between energy-cell and open-chamber engine

Fuel-Injection Equipment

It is a conceded fact that the fuel equipment can make or break a diesel engine.

A great deal of work, conducted by our Laboratory Engineers, was done in arriving at the proper fuel-injection equipment. It needed endless patience to make these tests, and, at times, we just had to "sweat it out."

Each type of combustion chamber has its idiosyncrasies, which require the patterning of the equipment to the job.

The pump, of American Bosch manufacture and design, is their model BB 6-cyl multiple-unit pump.

We arrived at the 9-mm plunger and the cam shape by the process of elimination. Larger plunger diameters or faster cams gave harsh operation. We always came back to the original conception that we wanted plunger velocity slow at the start and rapidly increasing subsequently. Before the fuel cutoff, plunger velocity should be reduced.

The delivery-valve design and its action, its displacer volume, as well as the pressure-line size, are of great importance.

The dead volumes have been reduced from 3896 cu mm to 1160. Fig. 31 shows two of the eight delivery valves that we tried.

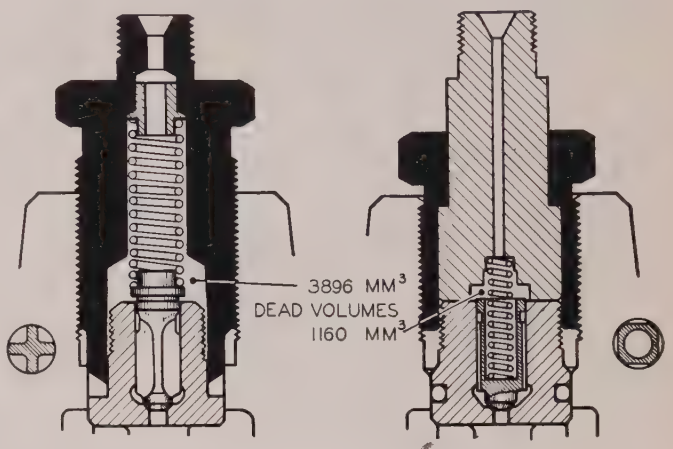


Fig. 31 – Delivery-valve designs

There is a difference in performance as can be seen from Fig. 32 in which bmep, smoke, and fuel economy are given. Too much volume in the delivery valve acts unfavorably upon injection. Some of the fuel is burned too late and is, therefore, of no value in producing brake mean effective pressure. Poor economy and smoke are the result.

Reverting to the valve, the design shown at the right in Fig. 31 has many advantages. The location of the spring and its size are significant. Its weight is less, and, because of its shortness and stiffness, the tendency to buckle and surge is reduced.

The length of the spring space can be held fairly accurate. It is not influenced by gasket compressi-

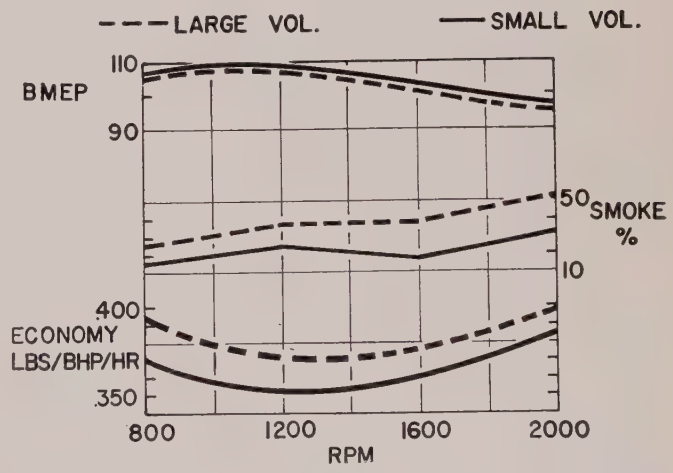


Fig. 32 – Effect of delivery-valve volumes on bmep, smoke, and economy

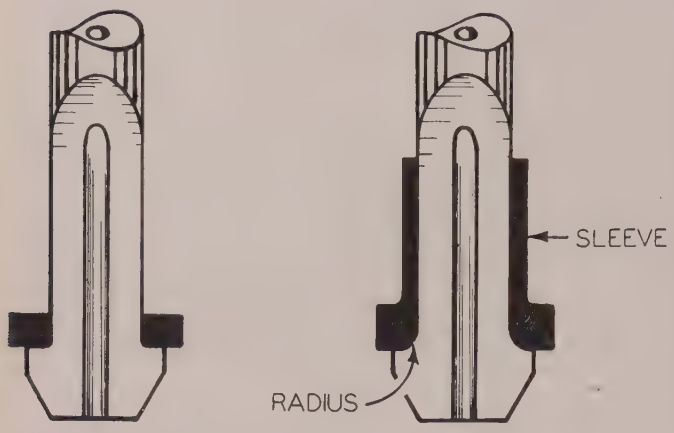


Fig. 33 – Conventional (left) and Mack (right) high-pressure fuel-line joint

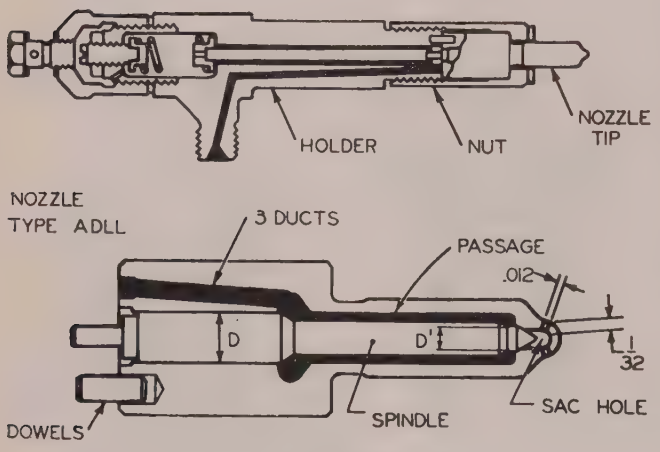


Fig. 34 – Nozzle assembly

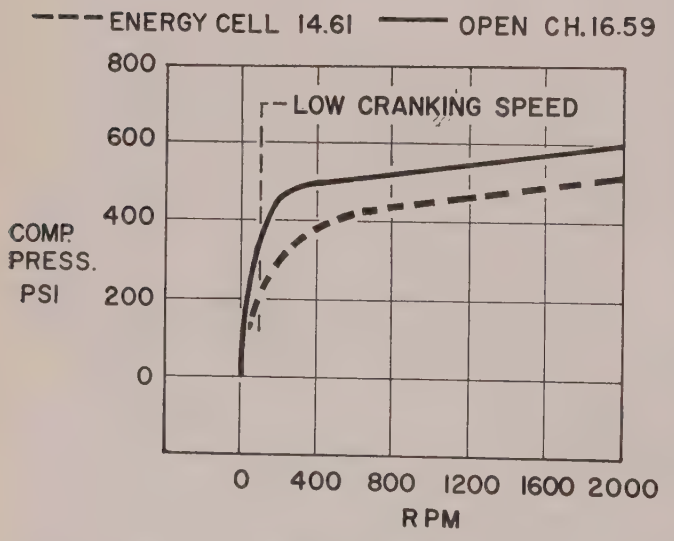


Fig. 35 – Comparison of compression pressures of energy cell versus open chamber

bility. Spring pressure can be held to closer limits, to help obtain uniform metering at idling.

An O-ring is used to seal against the low-pressure side. When tightening the delivery-valve body, the pressure passes through a solid column of steel. This prevents distortion of the body and eliminates valve sticking.

The bottom seat valve, because of its construction, has shock-absorbing qualities. Thus, it is quieter in operation. The guide vanes have more contact area while the grooving is arranged so that the valve does not spin.

Of importance is the fact that when tightening the delivery valve, no turning effort can scuff the lapped surfaces or bind the valve body. No "educated" wrench is required.

Next of importance are the fuel lines. These are equal in length, 35 in. The outside is ¼ in., the hole 0.068 in.

The line volume determines the displacer volume of the delivery valve. We tried delivery-valve displacer volumes of 50, 75, and 90 cu mm and found 75 to give, in our setup, the best economy.

After trials with various sizes, the 0.068 tube ID proved most suitable for our requirements.

Our pipe tube end design has been in use on all Mack diesel engines since 1938, and it is used on the open-chamber engine again. (See Fig. 33.)

Success in eliminating pipe-joint breakage can be attributed to this design, which shows a sleeve reinforcing the upset end.

The large fillet at the head in the Mack design contrasts sharply with the small fillet in the conventional type. Its benefits are manifest.

The nozzle consists of the holder and nozzle tip, as shown in Fig. 34. The holder was taken from the former energy-cell engine and modified to receive holes for the nozzle dowels.

The nozzle shown below in Fig. 34 is the ADLL type. It is used in most open-chamber engines in England and on the Continent.

There are many variables which we have tried. Variables were in the spray angle, number of holes, sizes of holes, L/d of hole length, and the differential area of the spindle, D'/D.

We found that the following combination gave us good economy and reduced knock. The specifications are:

Opening Pressure, psi	2300 ± 50
Valve, mm	6
Spindle, mm	4½
No. of Holes	4
Spray Angle, deg	150
Wall Thickness, mm	0.8
Hole Diameter, mm	0.325

Any other combination affected economy or smoke. This was especially true when more than four holes were used.

How sensitive some design modifications can become was demonstrated in this development. We used a nozzle with three duct holes, shown in the lower view. Because we had the nozzle dowelled, we thought that one duct hole, properly lined up

with holder duct, would improve the economy; it was just the opposite.

Increasing the nozzle spring pressure to 3100 psi or decreasing it as low as 1500 psi did not improve upon the performance of 2300 with which we started.

Starting Condition

Not only do we have a somewhat reduced combustion-chamber surface volume ratio in the open-chamber engine but also a greatly increased compression ratio. Fig. 35 gives the compression pressures for a 16.59/1 ratio. Of importance is the compression pressure at starting speed, that is, between 100 and 200 rpm. It is considerably higher than that of the energy-cell engine. Some good reports have come in concerning last winter's operation. Cold-room tests are now in process, but as yet results are not known.

It might be of interest that we started out with a 16/1 compression ratio. Subsequent changes on other than the combustion chamber raised it to today's figure, which agrees with prevailing practice both here and abroad. Maximum pressures are well under the expected figures. No trouble resulted on any of the bearings in 140,000 miles.

Volumetric Efficiency

Closely related to the compression ratio is the volumetric efficiency, as Fig. 36 shows. It can be seen that for essentially equal volumetric efficiency the air/fuel ratio is greater for the open chamber at equal bmep. This would indicate a higher safety factor against thermal overload.

Injection Advance

For automotive diesel engines that must have a wide speed range, it is beneficial to have an automatic injection advance. We have such a device incorporated in our engine, as shown in Fig. 37.

The mechanism consists of four fly-weights that move out. Weight arms move a collar axially; and by means of a left-hand and right-hand helix, the pump shaft is advanced. This mechanism is attached to the pump shaft, completely inclosed in the drive housing, and splash lubricated. The helix angle is such that no kick-back occurs. All parts are nitrided.

An Oldham coupling takes care of slight variations of the pump axis. The rear part of the coupling can be moved for correct pump timing. Once this is done, increased engine speed will automatically advance the pump.

On the energy-cell-equipped engines, we have had this advance, called synchrovance, for many years. It is of equal advantage on open-chamber engines.

In automotive diesels, automatic variation of injection timing is as essential to full efficiency as automatic spark advance is on gasoline engines. With fixed injection timing, sacrifice must be made either at the top end or bottom of the speed range.

Fig. 38 shows the improvement obtained by vari-

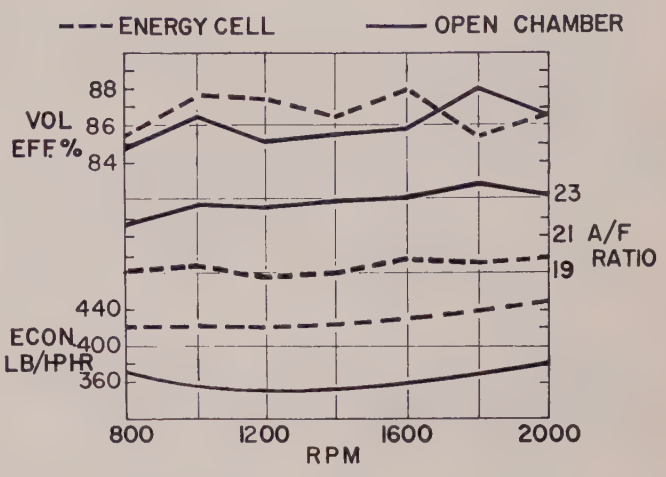


Fig. 36 – Volumetric efficiency, air/fuel ratio, and specific fuel consumption for open chamber as compared with energy cell

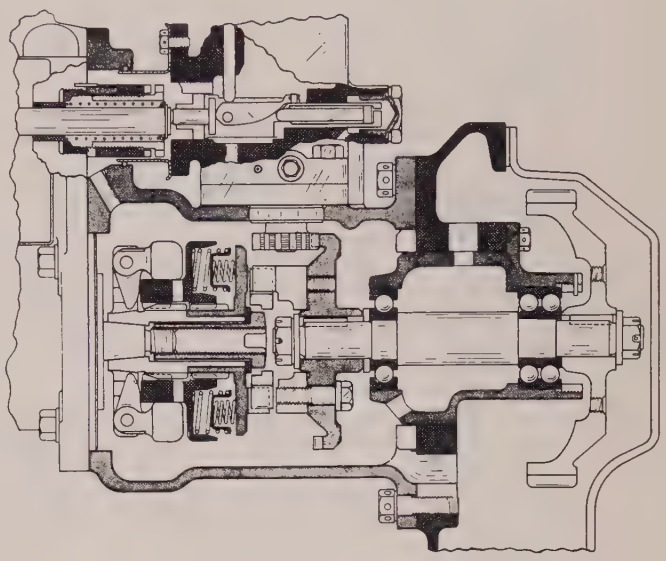


Fig. 37 – Mack synchrovance and governor drive

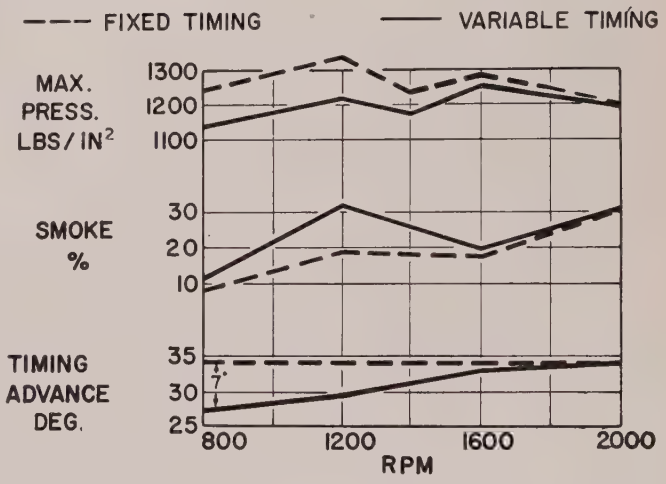


Fig. 38 – Effect of injection timing on maximum pressures and smoke

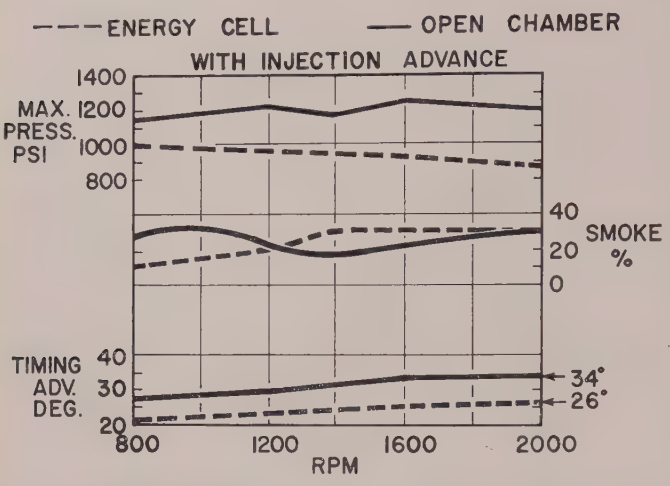


Fig. 39 – Comparison of maximum pressures between energy cell and open chamber with effect on smoke

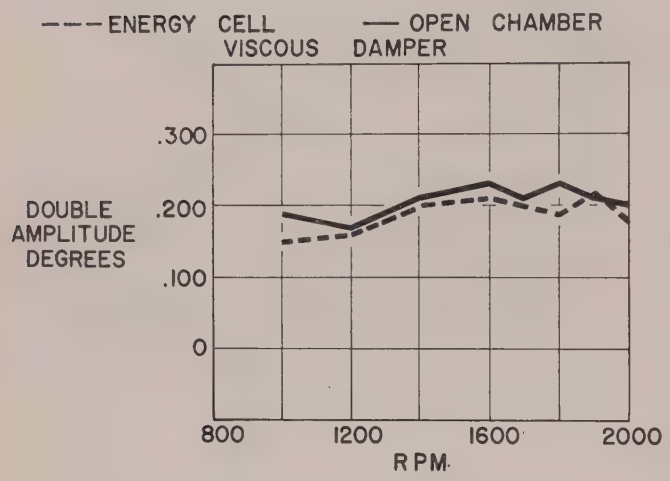


Fig. 40 – Comparison of torsional vibration

able injection timing in maximum cylinder pressures. The injection advance is 7 deg maximum. Thus, we have the most suitable injection timing for all speeds from 800 rpm up. The retardation at low speed reduces the engine knock and helps in cold starting.

It is interesting to compare the obtained maximum pressures of the open chamber with those of the energy cell. Fig. 39 gives this comparison.

Some people may say that this amount is high. However, we do not have any trouble resulting from this increase; and, when we consider the gain in economy, then, the balance is in favor of the open-chamber engine.

From this graph one can see the difference in the injection advance required between the open-chamber and the energy-cell type. Not only is the whole injection timing range earlier in the open-chamber engine, but the range of variable advance is less. This, of course, is due to reduced ignition lag. We attribute this to the direct versus the indirect injection, as used in the energy-type chamber.

Torsionals

Maximum pressures remind us of torsional vibration. Fig. 40 shows double amplitudes in degrees for the two combustion-chamber designs as applied to the same engine parts. Because of the increase in peak loading and the increase in piston weight, the open chamber has more amplitude. However, the viscous damper keeps the amplitude down to very reasonable figures.

Thermal Efficiency

The excellent thermal efficiency of the open chamber brings to mind the thought of radiator size because of the reduced heat rejection. Fig. 41 shows this condition in Btu per bhp per min. This graph and the following reflect the improved thermal efficiency over the older design. Fig. 42 indi-

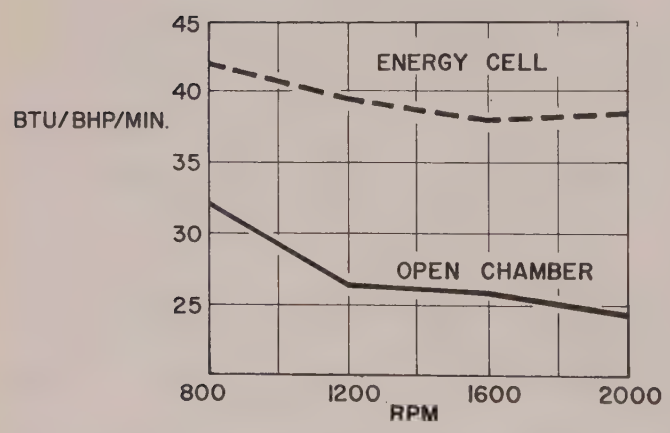


Fig. 41 – Heat rejection, Btu per bhp per min

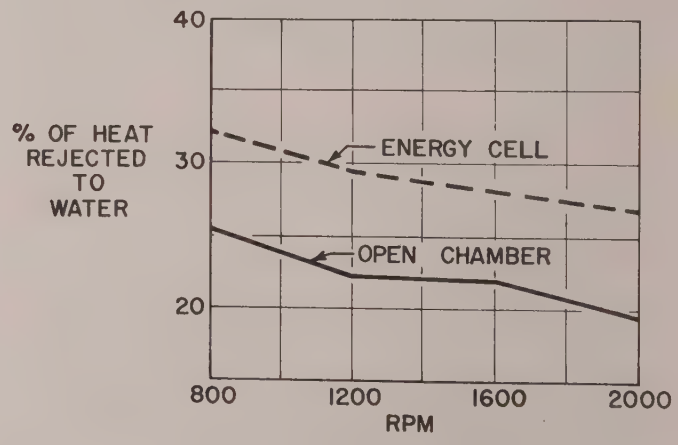


Fig. 42 – Heat rejected to water, %

Table 1 – Test-Fuel Specifications

	No. 2 Laboratory Fuel	No. 1 Kerosene
API Gravity	35.8	41.3
Pour Point	....[a]	....[a]
Initial Boiling Point, F	338	320
50% Distilled, F	510	445
Flash Point, F	162	131
Distillation Recovery, %	98	98
Cetane Number	50	49
Viscosity, SUS at 100 F	35.5	31.8
Water plus Sediment	....[a]	....[a]
Maximum Ash, %	0.002	0.001
Total Sulfur, %	0.24	0.08
Corrosive Sulfur (212 F)	....[a]	....[a]
Conradson Carbon on 10% Residuum	....[a]	....[a]
Residue	....[a]	....[a]
Mineral Acid, %	0.02	0.02

[a] Not checked.

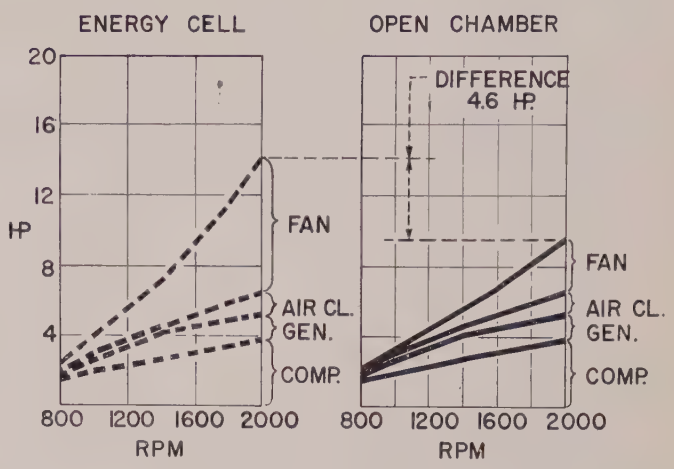


Fig. 43 – Accessory horsepower reduction with open-chamber design

cates that the water-pump capacity could be reduced. However, we prefer a copious flow and would rather reduce the radiator size where copper and weight could be saved.

Accessory horsepower on gasoline-powered trucks is high in comparison with the requirements of a diesel. However, Fig. 43 shows the difference in accessory horsepower between the two diesel-engine combustion principles. Naturally, the difference is put into the axle as useful power. This is another gain for the *net* horsepower of the engine.

Surface/Volume Ratio

Of interest is the surface/volume ratio, Fig. 44, between the two chamber designs. We thought that there would be a great difference in favor of the open-chamber design. Calculations, however, showed this to be much smaller than we anticipated, so we cannot give the lower surface/volume ratio much credit in improving the fuel economy. It must, therefore, be attributed to the excellent combustion as a result of swirl, squish, rate of injection, and spray pattern.

All our tests were run with No. 2 diesel fuel. It is important, however, for us to have some other grades of fuel tested. In the early stage of our development work, a test was run with No. 1 fuel. Fig. 45 shows that with equal pump setting, the bmep is somewhat lower, smoke is higher, and so is the specific fuel consumption. It is entirely feasible to compensate for the loss of bmep. However, the fuel consumption and smoke do not match that of the No. 2 fuel. Some of the experimental buses have covered up to 30,000 miles on No. 1 fuel; and after dismantling, combustion chambers were clean, and nozzles still functioned well. We do not anticipate any trouble from using this fuel in buses. Table 1 gives the specifications of the two fuels. Note that the sulfur content is below 0.5%. For lubricating oil to be used with these fuels, we recommend MIL-

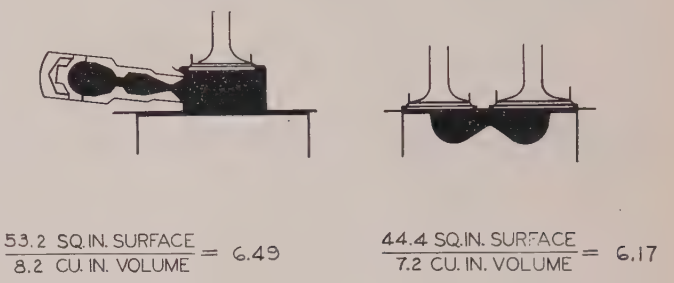


Fig. 44 – Surface/volume ratio for energy cell (left) and open chamber (right)

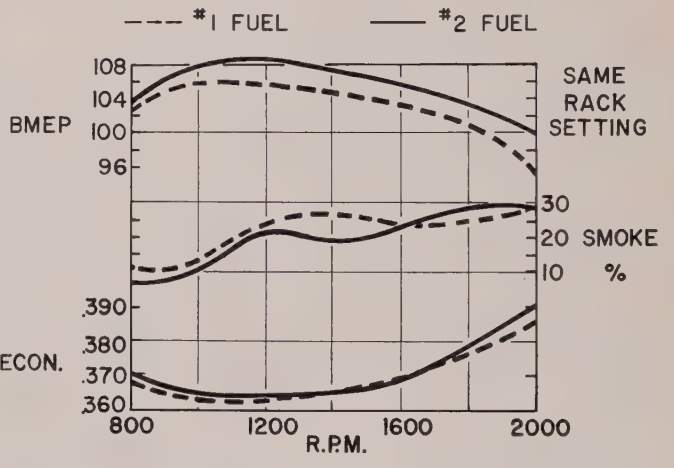


Fig. 45 – Effect on economy, bmep, and smoke with two different fuels at equal rack settings

0-2104. We have not as yet made tests with fuels having a high sulfur content. We would recommend, however, in the absence of our own data, a Series 2 oil.

Results

After the first test engines had shown encouraging results, we built several more engines for field tests, some for tractors, some for buses. Fuel and oil consumption were followed, and a careful record kept of each installation. Typical of the results was that of a tractor in regular highway freighting, whose gross combination weight was 57,300 lb and average trip speed 43 mph. Covering 186,159 miles, its total fuel consumption was 25,150 gal, representing 7.42 mpg.

On the basis of this evidence plus the highly satisfactory maintenance experience with these field-test engines, the new engine was placed in production last fall, and, in the year since, it has continued to show a degree of economy of both fuel and oil consistent with the test results in over 600 truck, tractor, and bus installations. Users' reports, carefully checked, show fuel mileages ranging from 6.44 to 8.00, according to operating conditions and loads for tractors, and from 4.35 to 6.2 in buses.

We are, therefore, satisfied that the operator can save from 15 to 20% of his fuel bill as compared with the best previous diesel experience, in addition to reduced maintenance expense.

Conclusion

We have shown you the development of our new open-chamber head and some of the results as compared with the existing combustion system.

There were many other interesting phases of this development which were not traced in this paper, but I feel that we have covered those of greatest importance. These alone are sufficient to explain the improvement in economy accomplished which, in turn, have justified the production release of the new design more than a year ago.

Credit for the development goes to our designers and laboratory engineers who have worked diligently on this project.

DISCUSSION

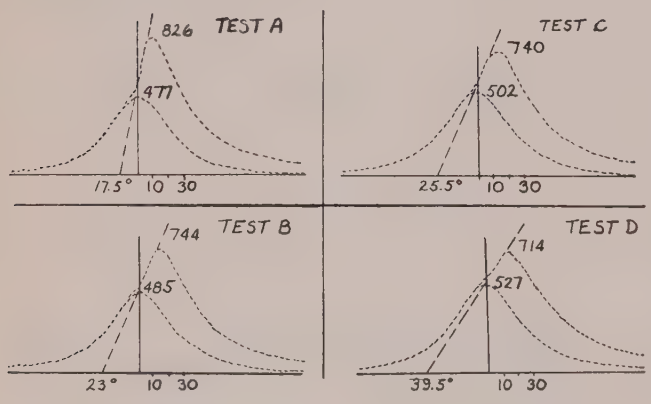


Fig. A – Indicator cards showing ideal combustion conditions – experimental 3¾ x 5 engine

	Test A	Test B	Test C	Test D
Speed, rpm	1200	1600	2000	2400
Beginning Injection, deg btdc	36	40.5	43.5	44.5
Pull, lb	28.8	27.6	26.2	24.7
Bmep, psi	103.8	99.4	94.4	89.0
Bhp	8.64	11.04	13.10	14.82
Fuel Rate, lb/hr	3.56	4.55	5.61	6.79
Specific Fuel Rate, lb/bhp-hr	0.412	0.411	0.429	0.458
Pressure Rise Rate, psi/deg	30.4	21.8	21.7	13.5
Ihp	10.89	14.44	17.95	21.60
Imep, psi	130.8	130.0	129.3	129.7
Specific Fuel Rate, lb/ihp-hr	0.327	0.315	0.312	0.315

Some Problems of Open-Chamber Engines

– Hans Fischer

Fischer Engineering

THIS paper gives valuable information on how to tune all the component parts to function for an efficient combustion. Especially gratifying to me is the fact that he gives due credit to the energy-cell combustion method of the single-lobe combustion chamber.

It is generally known that the open combustion chamber offers possibilities for the best fuel economy, but one has to sacrifice smoothness for it. Combustion approaches constant-volume burning rather than constant-pressure burning. The ideal combustion, but still with a high degree of fuel economy, requires burning about 60 to 80% of the fuel with the maximum pressure occurring at about 12 to 18 deg atdc of the piston. In order to accomplish such a combustion, the mixing of fuel and air has to be greatly accelerated after top firing center. The induced swirl and squish are most effective shortly before top center, but they are greatly diminished at top center and still more after top center when the formation of the mixture, and consequently the burning, has to be accelerated. Such combustion conditions are realized with the energy-cell combustion method.

The indicator cards as shown in Fig. A are an excellent example of ideal combustion conditions. The smooth start of ignition, the slow burning at top center, and the accelerated burning after top center are the outstanding features of the energy-cell combustion method. The rate of pressure rise is similar to that experienced with gasoline

engines. Please note also the indicated fuel rate, which is 0.315 lb per ihp-hr at an output of 129.7 psi mean indicated pressure at 2400 rpm. A comparative figure of an open combustion chamber is about 0.300 lb per ihp-hr, which is an improvement of about 5% only.

Combustion Chamber

Fig. B shows the latest single-lobe combustion chamber of the energy-cell type. The nozzle and cell are coaxial and diametrically across from one another. This arrangement resulted in better economy, cleaner exhaust, and easier cold starting. Fig. C shows the fuel economy at various speeds of a tractor engine with the improved chamber. For an engine with an auxiliary chamber, the fuel consumption is outstanding, if one considers that the engine was equipped with radiator, fan, generator, and air cleaner.

It is generally recognized that the specification of the fuel for the open combustion chamber is much more narrow than for an engine with an auxiliary chamber. Such fuel is more expensive.

The requirements on the injection system for diesels with main and auxiliary combustion chambers are more easily met than for those with open combustion chambers. Generally lower injection pressure is required, which means less expensive injection equipment but also less wear and tear on the pump.

I have experienced greater difficulties in starting a well-known small open-chamber engine around freezing temperatures than with the latest energy-cell-type engine.

Engines with an auxiliary chamber can be idled for any length of time, but not so with an open-chamber engine with which one can get into serious trouble with nozzles and pistons if the engine idles for a long period of time.

The application of the energy-cell-type combustion method is limited to small and medium-size automotive engines. Mr. Loeffler's engine with a 4⅞-in. bore, I consider already too large for efficient energy-cell action.

Chamber Surface in Piston Improves Fuel Economy

– John B. Reeves, Jr.

International Harvester Co.

THIS paper shows us what real accomplishments can be made by a complete review of competitive equipment, intelligent design and analysis, and tedious but careful development. Mack must be congratulated for their foresight and initiative, which have resulted in a truly highly efficient automotive diesel engine that compares favorably with those produced by our European competitors.

Should other American diesel-engine manufacturers, particularly those who use indirect injection, attempt to compare their engines to the new Mack engine, they will find that the brake thermal efficiency of the Mack system far surpasses their own. International Harvester engines, for example, are of the precombustion-chamber design.

Our diesel engines are designed for industrial power applications, therefore a direct comparison cannot be made readily on the basis of brake horsepower and fuel consumption. It is still worthwhile, however, to review a few performance figures.

Our smallest diesel, 264 cu in., and our largest, 1091 cu in., both deliver 110 bmep at their maximum torque points on the stripped intermittent power curve with a fuel consumption of 0.420 lb per bhp-hr. This compares favorably with the consumption of the Mack-Lanova system but is a long way from the 0.360 at 110 bmep obtained with the improved Mack combustion system.

Further comparison would be superfluous because of the dissimilarity of the two combustion systems. These figures

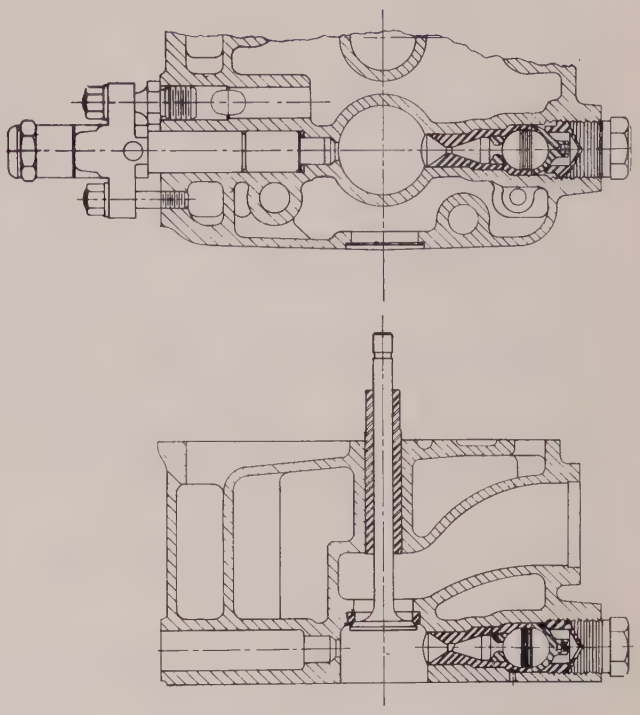


Fig. B – Single-lobe combustion chamber

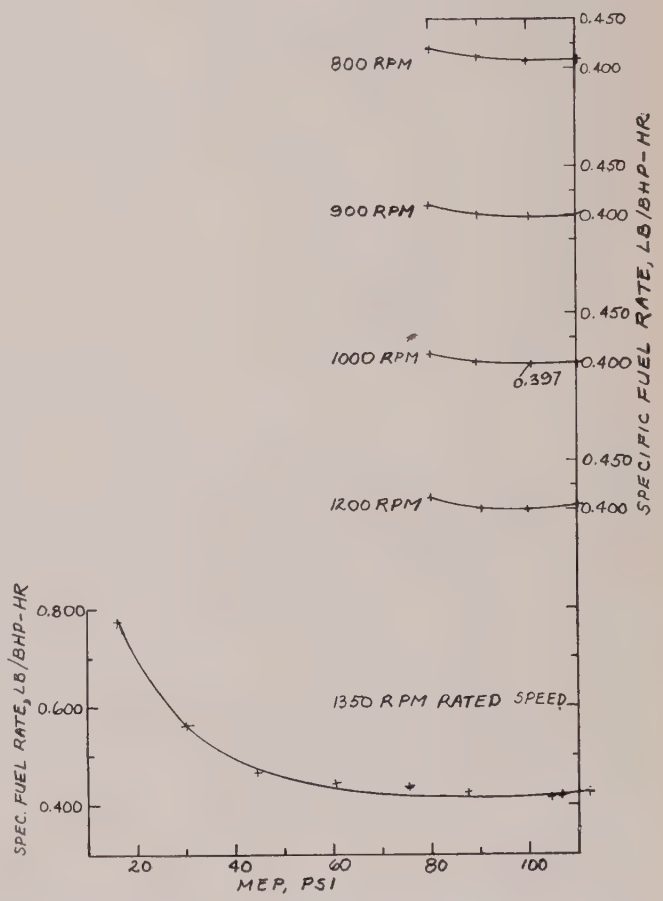


Fig. C – Fuel economy and performance of 4-cyl tractor engine (251.4 cu in. displacement) with improved combustion chamber – engine equipped with radiator, fan, generator, and air cleaner

are comparable to many competitive engines and indicate one important consideration. Combustion systems as designed and manufactured in this country, although capable of producing excellent thermal efficiency, can be further improved by an intensified combustion development program. In certain cases, this improvement will be accomplished at the expense of rather simple injection systems.

The data presented by Mr. Loeffler are indeed complete. I would like to ask him a few questions concerning some interesting points.

I have assumed that the performance data presented is for stripped engines. Since friction horsepower has a major effect on brake specific fuel consumption, it would be of interest to know what difference exists between the friction horsepower of the two combustion systems. It is highly possible that reduced friction due to reduced pumping losses was nullified by increased compression ratio.

The characteristics of the potential power curves shown in Fig. 30 indicate an appreciable gain in absolute maximum bmep for the two speeds given. This is also indicated by the air/fuel ratios given in Fig. 36. It would be interesting to know if there was an appreciable gain, and, if so, was it due to reduced friction horsepower or entirely due to improved combustion.

With regard to starting characteristics, is there additional data available at this time? In particular, what is the lowest temperature at which the engine will start without aid?

The unequal delivery through the spray holes is another interesting point. I wonder if Mr. Loeffler can advance a theory as to the cause of this unequal delivery?

Mr. Loeffler noted that the surface/volume ratio changed only slightly, and he could not give the lower ratio much credit in improving the fuel consumption. It should be noted that because of the configuration of the old combustion chamber most of the surface was in the cylinder head and, therefore, relatively close to the coolant chambers. In the new combustion chamber, the greater portion of the surface is in the piston. Therefore, it appears to me that the relocation of the combustion-chamber surface contributes to the lower heat rejection and improved fuel economy.

Automatic Injection Advance Not Needed On All Diesels

– W. A. Wallace

Caterpillar Tractor Co.

THE author has given us an excellent discussion of the many problems associated with the development of a new combustion system for a diesel engine. Instrumentation in engine research has been greatly improved during the past few years; however, combustion-system development still remains pretty much a cut-and-try proposition. Thus, the development of a new combustion system is a long and tedious job, since the variation in design of the component parts is almost limitless.

This paper covers many phases in the development of a given combustion system. I would like to comment briefly on a few items in which my company's experience has differed somewhat from that of the author.

In the discussion of the design of the inclined fuel injection nozzle, emphasis was placed on the fact that the two spray holes on the side pointing toward the more remote side of the chamber delivers about 2% more fuel than the other two. I am wondering if this variation in discharge were actually measured, and, if so, whether it were consistent. If one were to guess whether one orifice would have more discharge than another, it would be natural to choose one of the two orifices on the side pointing toward the more remote side of the chamber to have the greatest discharge, since the restriction to fuel flow in this direction should be less. However, a 2% variation in discharge between four orifices of 0.013 diameter would probably be considered very satisfactory from a production standpoint regardless of the location of the orifices in the nozzle.

Reference was made to the fact that valve recesses in the cylinder liner had an adverse effect on fuel economy. I would agree that recesses in the cylinder liner should be avoided if at all possible from the production standpoint, but I am wondering whether the author has definite proof that these recesses actually influence fuel consumption. Off-hand, it would seem that the effect on fuel consumption should be the same, regardless of whether the recesses were located in the liner and piston or entirely in the piston.

During the discussion of the piston-and-ring design, the author stated that the top-groove temperature of the direct injection piston was lower than that of the Lanova combustion-chamber piston. During the past few years, many observations of top-groove temperatures have been made by the Research Department of Caterpillar Tractor Co. on pistons used in various combustion-system designs. In making these observations, instrumentation was used that allowed a continuous recording of piston temperatures to be made under various operating conditions of the engine.[a] At no time have we been able to show that top-groove temperatures in pistons of direct-injection engines are any lower than those of pistons used in precombustion-chamber engines under identical engine operating conditions. If the author has made actual temperature measurements in the Lanova and direct-injection engines, it would be interesting to know what method he used for obtaining the temperatures and how much variation in top-groove temperatures he measured. Along this same line, reference was made to the fact that ring sticking was greatly reduced in the direct-injection engine through the use of the Keystone ring design. No information was given on the recommended oil-change periods to be used on this new engine. It would appear possible that the oil-change periods for this new engine have been reduced from that previously recommended for the Lanova chamber engines. Perhaps the author could give us a little more information on this comparison of the two engines. Also, I wonder whether the oil-change period for the direct-injection engine is determined by a viscosity increase of the lubricating oil or by the depletion of the additive content of the oil.

The need for automatic injection advance on automotive diesel engines that have a wide speed range is a controversial point. For some types of diesel engines, it is necessary to advance injection timing with an increase in speed to obtain optimum performance. However, I do not believe that automatic variation in injection timing for all automotive diesels is as essential to full efficiency as automatic spark advance is on gasoline engines. We have not been able to show any appreciable improvement in efficiency due to variable injection timing on our standard precombustion-chamber diesel engines that operate over a 2/1 full-load speed range with a top speed of 1900 rpm. This characteristic has also been observed on an experimental small-bore precombustion-chamber diesel engine that operated over a 2/1 full-load speed range with a top speed of 4800 rpm. On the engine described by the author, automatic injection advance with speed may be beneficial, but I seriously doubt if it is a requirement for all automotive engines.

The performance curves of the engine show that the author accomplished his initial purpose in building an engine that had very attractive fuel consumption characteristics. Also the very low total heat rejection of the open-chamber engine, as shown in Fig. 41, is quite outstanding.

[a] See Proceedings of the Society for Experimental Stress Analysis, Vol. IV, No. 2, 1947, pp. 52-61: "Device for Maintaining Continuous Electrical Connections with Reciprocating-Engine Parts," by W. A. Wallace and W. A. Casler.

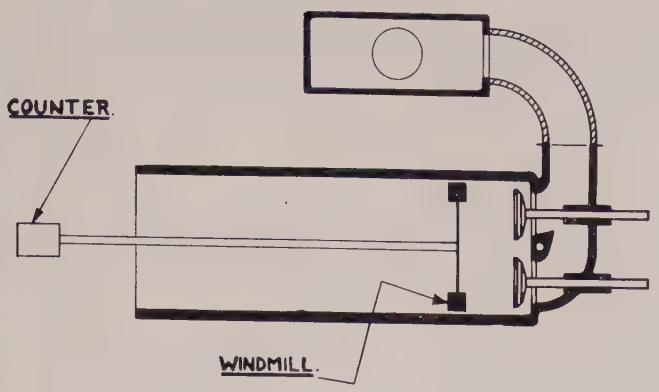


Fig. D – Schematic diagram of static setup simulating in-line engine

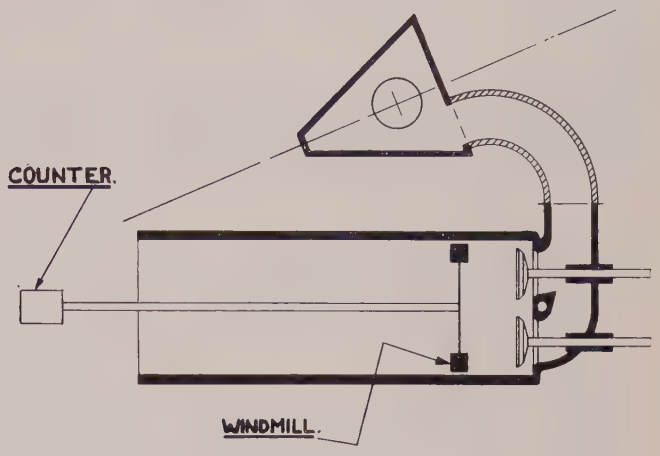


Fig. E – Schematic diagram of setup corresponding to 45-deg V engine

This certainly provides an easy method of reducing accessory horsepower and making more power available for useful work. Considerable emphasis was also placed on obtaining smoke-free operation to as great an extent as possible. This I believe is very commendable since, as the author stated, the general public is becoming more and more indignant over the increasing smoke nuisance caused by the majority of diesel engines in commercial vehicles. However, I do not concur that smokeless operation necessarily means moderate exhaust temperatures nor that lower exhaust temperatures necessarily means more complete combustion occurring in the cylinder. Depending upon combustion-chamber design and the degree of development, it is entirely possible to get smoky operation with very moderate exhaust temperatures if mixing between the fuel and air is poor. It takes a relatively small portion of the total fuel burned in an engine cycle to cause a smoky exhaust, and I seriously doubt whether exhaust temperature can in any way be correlated with smoke-free operation.

Comments on Swirl And Fuel Injection

– M. Berlyn

American Locomotive Co.

THE two areas of this development on which I am going to comment are (1) the fuel injection system and (2) the means of producing swirl in the air charge.

I agree emphatically with Mr. Loeffler's remarks on the value of minimal dead volume on the discharge side of the fuel-pump delivery valve; his observations on pump plunger velocity are very interesting, but I would like to know if he has had trouble with erosion of the tubing bores on this engine after prolonged operation.

The nozzle specification is surprising, particularly the ratio of the differential diameters of the valve. If the opening pressure setting is 2300 psi and if the differential diameters are in the ratio of 4 to 3, as Mr. Loeffler gives them, then the nozzle closing pressure is 1020 psi, considerably below the cylinder firing pressure of 1200 psi. I would have expected hot-gas blowback into the nozzle with this combination.

In the work of Ricardo on air swirl, referred to by Mr. Loeffler, the air rpm/crank rpm ratio between nine and ten was referred to a combustion chamber employing a single-orifice nozzle; we have found that a much lower swirl rate is best with multiorifice nozzles.

Now with regard to the development of the optimum air swirl in the cylinder of the Mack END673, I am surprised by Mr. Loeffler's implication that this was carried out on a 6-cyl engine and not on a single-cylinder job.

In the investigations of air-swirl-producing means which we have carried out in our laboratory, our routine has been, first, to observe swirl characteristics of a wood-model cylinder head bolted to a liner and fed with shop air under arbitrarily standardized conditions; then we run similar tests on an iron head to verify that a fair reproduction of the wood model has been achieved; after that, we run the iron head on a single-cylinder engine; and finally we run a whole set of them on a multicylinder engine.

We have observed that the way in which the swirl is produced influences the value of swirl rpm giving optimum results in the engine. In general we need to see nearly twice as many swirl rpm on our swirlmeter when using port-directed swirl as when creating the swirl with masked valves. We suspect that this is because masked valves give vigorous swirl even at partial opening, whereas the port-directed swirl is weak until the valves are almost fully lifted.

We have also found that port-directed swirl can be very sensitive to air manifold and elbow configuration.

We have a condition requiring that a cylinder head will give the same swirl characteristics whether employed on an in-line engine or a V-type engine. With port-directed swirl this is not easy.

Fig. D shows schematically our static setup simulating an in-line engine; it includes a 90-deg elbow connecting the air manifold and the port in the head.

Fig. E shows a similar setup corresponding with our 45-deg engines, this one incorporates a 112½-deg elbow between the air manifold and the port in the head.

These two setups have given vastly different swirl results, although the flow was almost identical in each case.

But that is not the worst. In our V engines the air manifold is of the dead-end type; the air from the supercharger enters at one end only. In effect, therefore, the air approaches the right-bank heads from one side and the left-bank heads from the other, as shown in Fig. F.

To determine the influence of this condition on swirl and flow in the left-bank and right-bank cylinders we took the setup as in Fig. E and fed the manifold section first

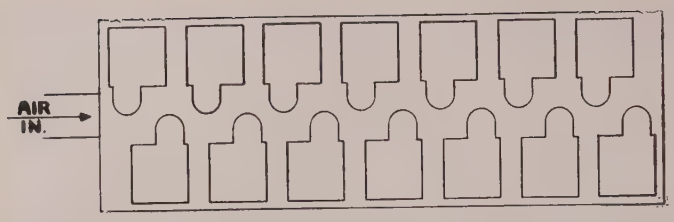


Fig. F – Dead-end type of air manifold

at one end and then at the other, making the cylinder head think, first, that it was a left-bank head, then a right-bank head.

The swirl differed in a ratio of more than 2/1, and the flow was about 10% higher in the low-swirl case.

Now the Mack END673 is not a V-type engine; but Fig. G, a rough representation of the Mack air manifold, makes one wonder if there may not be at least two different kinds of swirl in this engine, one kind in cylinders 1, 2, and 3 and another kind in cylinders 4, 5, and 6.

It should be emphasized that the effects I have just reported were observed with four-valve heads having two air valves. We recently ran a check on a two-valve head design having a single air valve, and found it very much less sensitive to what we call "elbow effect" than the four-valve heads which we use. Unfortunately, this two-valve head has limitations, unrelated to swirl, which we have not yet overcome.

Describes Saurer Dual-Turbulence Diesel

– Société Anonyme Adolphe Saurer

THE author's analysis and comparison of the numerous direct-injection combustion systems as used in Europe can no doubt be regarded as a very useful contribution towards the improvement and furtherance of the economical aspect of the diesel engine for road transportation.

May we, however, here be permitted to register the fact that the "originators" of the direct-injection system utilizing inlet and compression air swirls to improve combustion at higher engine speeds were the Saurer Co. (commercial vehicle manufacturers) of Arbon, Switzerland.

This company already in 1928 brought out a 4-cyl road diesel engine. The combustion system then employed was the "ACRO" air-cell type, which was superseded around 1930-1931 by an improved arrangement, the so-called cross-flow system. Whereas with the latter system engine output and flexibility were quite satisfactory, the fuel consumption was still rather high.

Considerations identical and of the same order and importance as Mr. Loeffler puts forward in his paper led the engineers of the Saurer Co. around 1933-1934 to study ways and means to combine the advantages (good flexibility) of the precombustion-chamber types of engines with those of the slow-running stationary engines or ship diesel engines, with their phenomenal low fuel consumption.

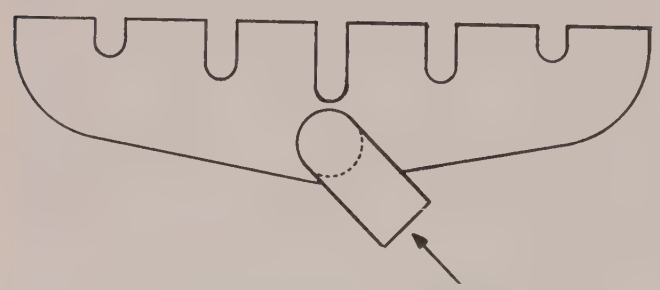


Fig. G – Rough representation of Mack air manifold

After extensive research and test work, going practically through all the stages of difficulties also encountered by the author with his new engine, the Saurer dual-turbulence deisel engine was put in production and operation in 1934.

The Saurer dual-turbulence diesel engine, described in many technical papers and journals during the past 20 years, employed a heart-shaped cavity in the piston, similar to the example F as shown in Fig. 2 of this paper. Four valves of equal diameter were used per cylinder. The two inlet valves carried shrouds and were so positioned as to give an intensive induction swirl horizontally around the vertical cylinder axis, even at low engine speed. Due to the special form of the chamber in the piston head, a supplementary air-swirl, in the vertical sense, is created during the compression stroke, the intensity of which increases very much owing to the squish action as the piston approaches the end of its upstroke. A 4-hole fuel injector is placed centrally between the valves in the cylinder head, and is located in a water-cooled copper tube.

The outstanding performance of the first Saurer dual-turbulence engines found greatest interest with other engine makers, some of which obtained the manufacturing rights of this combustion system by license. Further, the excellent performance figures obtained with this system induced subsequently more and more of the leading European road-diesel-engine manufacturers, especially in England, to divert from the earlier precombustion-chamber designs to a new direct-injection arrangement.

For Saurer's as pioneers in the design of direct-injection combustion systems it is somewhat gratifying to note from the author's analysis of direct-injection systems that practically all of these have some things in common with the fundamentals of the direct-injection dual-turbulence system of Saurer. It may also be of interest to mention here that this dual-turbulence system, first introduced some 20 years ago, is, with the exception of some newer refinements, still being employed with all Saurer 4-stroke diesel engines produced today.

In reviewing the great variety of direct-injection systems already existing and considering the forms and proportion which in some cases vary so much from one another, it is very surprising that the results published from all these different designs can yet generally be regarded as good. The present adoption of this direct-injection system in the United States appears to be a new confirmation of the superiority of this system over all other designs.

Referring to the new design as proposed by the author, it would appear that the open, wide, offset cavity in the piston must have given a fair amount of trouble until the correct proportion was found for the given bore and stroke measurements. There is, further, no doubt that with this wide chamber most minute studies of the fuel-injection arrangement were needed. The required long-way penetration of the fuel sprays demands a very high delivery speed, high injection pressures, and a most careful design and execution of the fuel injection nozzle. In order to obtain smoke-free combustion at engine speeds of 2000 rpm, the injection advance must be set very early before top dead-center. This will, however, cause heavy diesel knock at lower engine speeds. An automatic device controlling the fuel injection advance relative to the engine speed becomes therefore indispensable. Also, due to the large diameter of the combustion chamber, the compression swirl and squish action is of low order, as confirmed by the author. This necessitates the inducement of an induction swirl of utmost intensity to assure a certain swirl energy also at low engine speeds. This, in turn, demands high air speed in the induction line, which by nature causes losses and reduction in volumetric efficiency.

When the author has succeeded mastering all these intricate problems with good compromises and especially with the ingenious form of the induction port as described, he is certainly to be congratulated on his great achievement.

Development of the Turbo Compound Engine

F. J. Wiegand and W. R. Eichberg
Wright Aeronautical Division, Curtiss-Wright Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1954.

THE Wright Turbo Compound is an 18-cyl double-row aircooled radial engine having a piston displacement of 3350 cu in. The basic engine represents a continuous improvement in the design and performance of the original engine, which powered the B29's during World War II. In addition to increasing the power output within the cylinders, engine efficiency and output have been further increased by means of turbine compounding. This has been accomplished by passing the engine exhaust gases through three blowdown turbines mounted directly on the engine, which in turn feed the recovered power back into the engine crankshaft. The turbines do not require any new controlling element nor impose any operating limitations on the engine. Other than the required turbine exhaust hoods, the exhaust system has been made an integral part of the engine.

A brief summary of the horsepower advances of the R-3350 engine, together with the accompanying weight changes, may be seen in Fig. 1. It will be noted that the consistent reduction in the weight per horsepower ratio continued with the addition of compounding.

Initial design study from the standpoint of permissible turbine size and desirable exhaust pipe

THE 18-cyl turbine-compounded engine developed by Wright is claimed to be an attractive powerplant for aircraft designed to fly up to speeds of about 450 mph.

The compounding of this engine is accomplished by means of three blowdown turbines. This type of turbine is used because, according to the authors, it is more efficient than the pressure turbine up to an altitude of about 30,000 ft. It is estimated that at take-off a suitable pressure turbine would provide about 8% increased engine output. The present Turbo Compound engine, with its blowdown turbines, gives an 18% increase.

The Authors

F. J. WIEGAND (M '37) is director of rotating-engine development for Wright Aeronautical Division, Curtiss-Wright Corp. He attended Yale University and received a B.S. degree in mechanical engineering in 1933. During World War II, Mr. Wiegand served as a ground school instructor for the C.A.A. and during his professional career he has been active in A.I.A. – ETC and S.A.E. committee work.

W. R. EICHBERG is manager of the Reciprocating-Engine Division, Wright Aeronautical Division, Curtiss-Wright Corp. He received his B.S. degree in mechanical engineering in 1942 from Purdue University. Mr. Eichberg joined Wright as a junior test engineer in 1942.

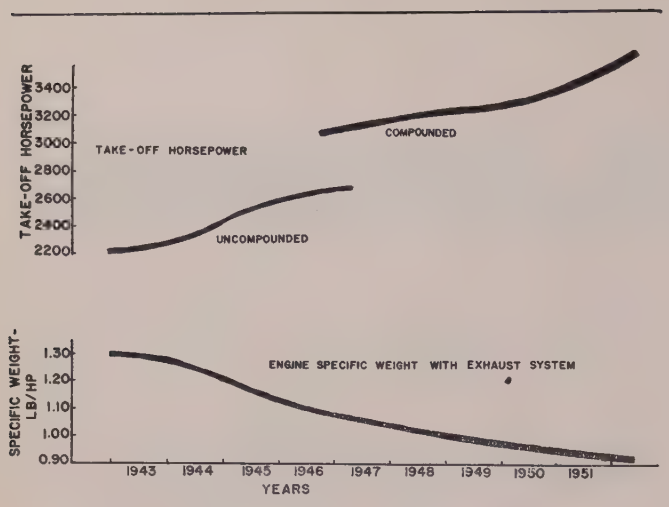


Fig. 1 – Development growth of Wright R-3350 engine

configuration favored the use of six equally spaced blowdown turbines, as shown in Fig. 2. Each turbine nozzle was divided into three separate sectors, permitting an individual exhaust pipe for each cylinder, thereby eliminating any adverse exhaust gas scavenging conditions that might exist if two or more cylinders were routed to a common turbine inlet pipe. Such an arrangement was also considered desirable from the standpoint of turbine blade durability, for it would permit optimum blade cooling as a result of the longer period between power impulses.

However, the possibility of providing three turbines by siamesing two cylinders into a common turbine inlet pipe was also considered. Such an arrangement presented a substantial weight saving. The selected configuration is shown in Fig. 3. The exhaust system is identical for each turbine location, thereby simplifying exhaust system design and minimizing manufacturing costs. Two adjacent cylinders of each row are siamesed together, and since the period during which the exhaust valves are open simultaneously is very small, this arrangement did not present a cylinder scavenging problem. In this system the remaining two cylinders that are interconnected comprise an adjacent front and rear cylinder. Here, both exhaust valves are open together for a period of 95 deg of crankshaft travel. The front-row cylinder is completing its exhaust cycle as the exhaust valve of the rear cylinder is opening, thereby presenting the possibility of backflow of exhaust gas into the front cylinder. This condition upsets proper cylinder scavenging and impairs engine performance. As shown in Fig. 4, turbine inlet pipes were specifically designed to minimize the backflow of exhaust gas. This was accomplished by extending the leg of the rear-cylinder exhaust pipe down stream within the pipe itself. The exhaust gas impulse flowing from this pipe must completely reverse its direction to travel back down the adjoining leg. This arrangement also permits the gas flow from one cylinder to aspirate the connecting cylinder, further improving scavenging.

Early engine development testing showed that greater power recovery could actually be realized when the number of turbines were reduced from

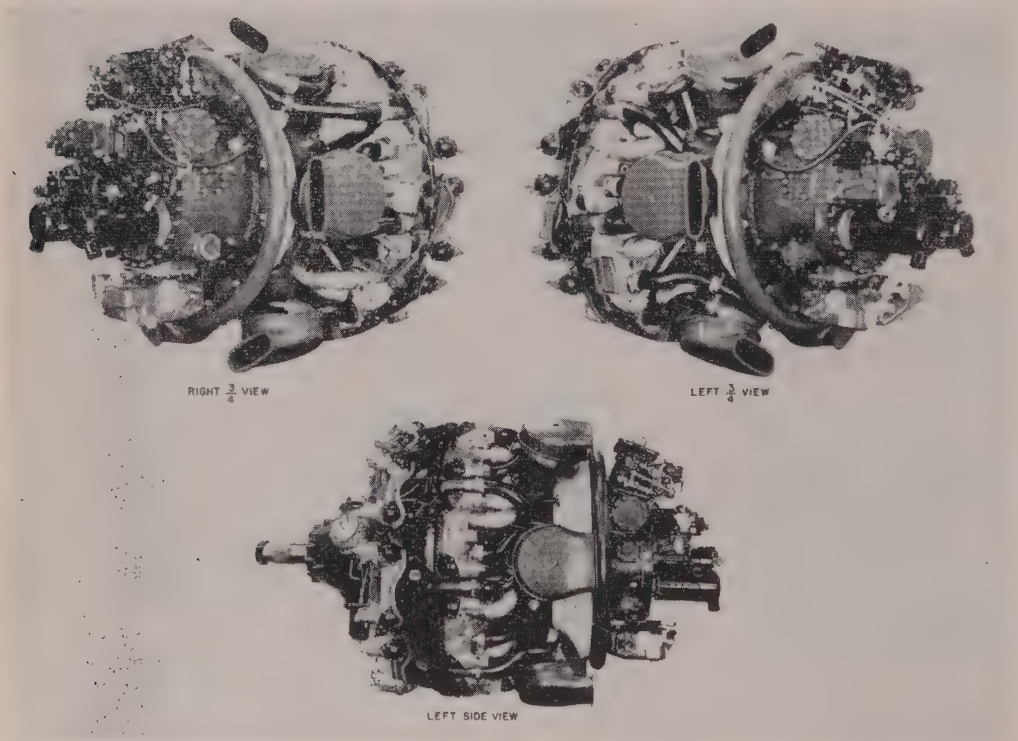


Fig. 2 – Early prototype model – 6-turbine engine

six to three. This was attributed to an overall reduction in friction, cooling air pumping, and turbine wheel windage losses.

As a result of improved turbine performance, satisfactory turbine durability, and the accompanying weight reduction, the 3-turbine arrangement was selected for the production engine.

The total engine development time for the R-3350 engine since 1944 is shown in Fig. 5. As indicated by 10,000 hr of development testing, considerable basic engine background existed prior to the addition of the turbines. In October, 1949, the engine completed its 50-hr official flight approval test. In January, 1950, the 150-hr official Navy qualification test was completed, and shipment of the first production engines started in March, 1950. Development of the Turbo Compound engine since that time has continued with sizable power increases in both the take-off and maximum cruise ratings. Engine operation has been improved and fuel consumption has been further decreased by the addition of direct cylinder head fuel injection.

Substantial quantities of these engines have been manufactured to date and these engines are installed in airplanes engaged in patrol action, troop and cargo transport, and commercial airline transportation. Commercial airliners using the Turbo Compound engine are the Lockheed P2V, Martin P5M, Fairchild C-119, the Lockheed Constellation, and Douglas DC-7, shown in Figs. 6 through 10.

Description of Theory

It is desirable from a thermodynamic standpoint to keep the exhaust valve closed during the expansion stroke until complete expansion of the gas within the cylinder has occurred, thereby obtaining maximum work output. There are, of course, practical limitations to the degree of expansion permissible within the cylinder. Such factors as engine size, piston stroke, permissible piston speeds, pumping losses, and heat losses, must be considered together with the necessity of rapidly scavenging the cylinder prior to the beginning of the next cycle. As a result of these considerations the exhaust gases upon being released from the cylinder through the exhaust port still contain considerable energy that can be converted to useful work.

This condition has been generally realized in the internal-combustion-engine field, and several systems of power recovery have been developed and applied to aircraft installations. As shown in Fig. 11, these may exist in the form of jet thrust or in the form of turbine power recovery. The simple collector ring will develop some thrust if the outlet is pointed rearward; however, sizable losses are incurred as the exhaust gases leaving the cylinder expand abruptly into the large manifold. Power recovery at moderate air speed has been estimated to be about 3.0% of the engine output. This condition can be improved by providing jet stacks permitting direct conversion of exhaust gas velocity energy into thrust. Power recovery in this case

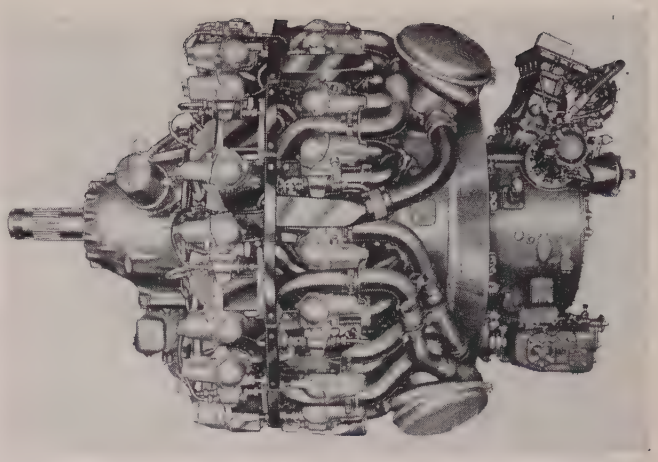


Fig. 3 – Final production model – 3-turbine engine

Fig. 4 – Aspirated turbine inlet pipe

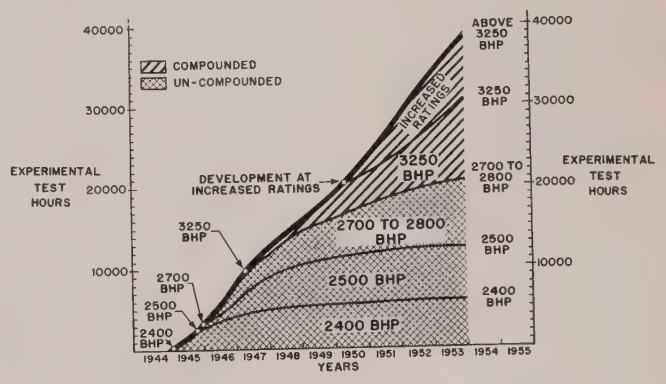


Fig. 5 – C18 engine development curves – experimental test hours

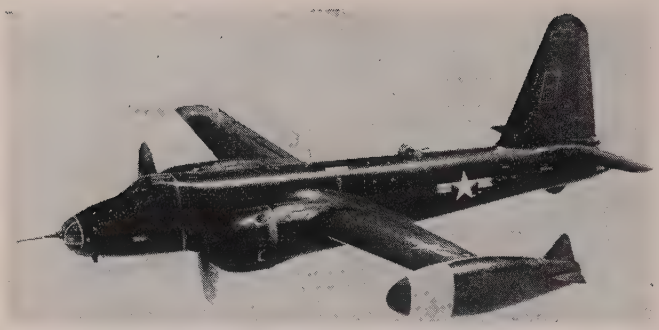


Fig. 6 – Lockheed P2V airplane

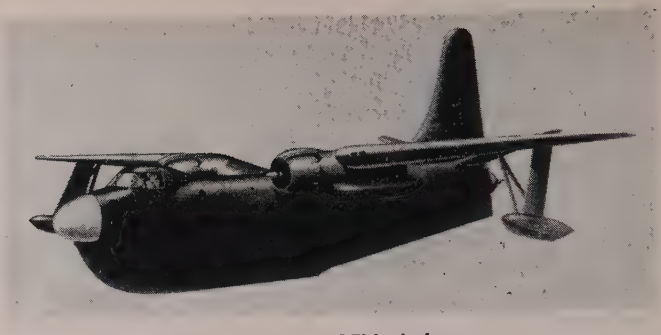


Fig. 7 – Martin P5M airplane

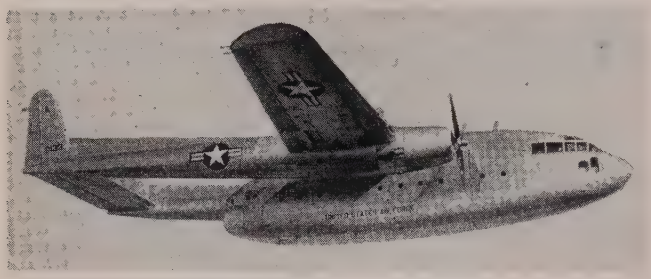


Fig. 8 – Fairchild C-119 airplane

Fig. 9 – Lockheed 1049 Super Connie

Fig. 10 – Douglas DC-7 airplane

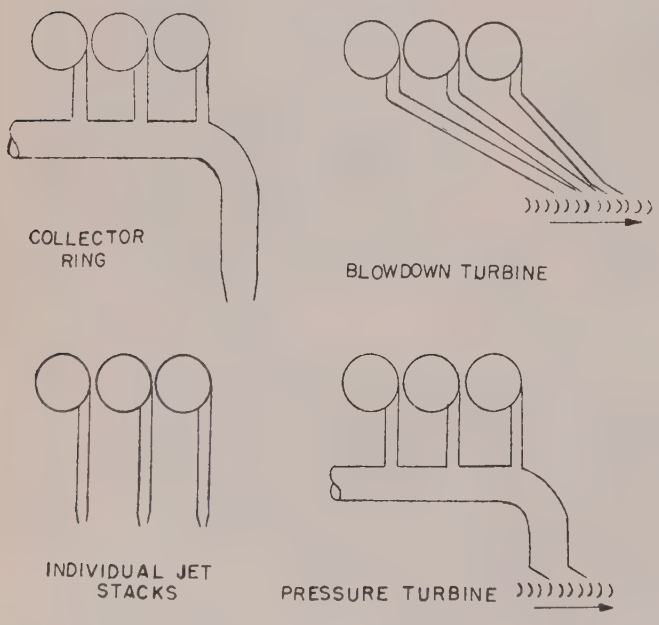


Fig. 11 – Power recovery systems

will be increased to approximately 10.0% of the power output.

There are two principal methods of recovering exhaust gas energy through the use of a turbine or turbines. The impulse or blowdown turbine utilizes the kinetic or velocity energy of the gas as it is deflected through the turbine wheel blading. The pressure turbine receives its energy from the expansion of gases in the nozzle and turbine blades. This system requires a pressure buildup of the gases within a collector manifold and is obtained by reducing the nozzle discharge area. Study has shown the blowdown turbine to be more efficient than the pressure turbine up to approximately 30,000-ft altitude.

The efficiency of the pressure turbine is largely dependent upon the ratio of the exhaust collector manifold pressure to turbine exhaust pressure, which at sea level cannot be too large because of engine exhaust back pressure considerations. As the exhaust pressure decreases with increased altitude this pressure ratio increases and turbine performance improves. Power recovered from the blowdown turbine also increases with increased altitude as the exhaust gases, upon being released from the cylinder, "blowdown" to a lower atmospheric pressure. The increase in gas velocity, however, is partially offset by increased friction losses within the turbine inlet pipes. At sea level, therefore, power recovered with the blowdown turbine exceeds that of the pressure turbine and, although the output of the pressure turbine increases with altitude at a greater rate and will gradually out perform the blowdown turbine, this will not occur within the range of altitude obtained with the Turbo Compound engine. Due to the nature of

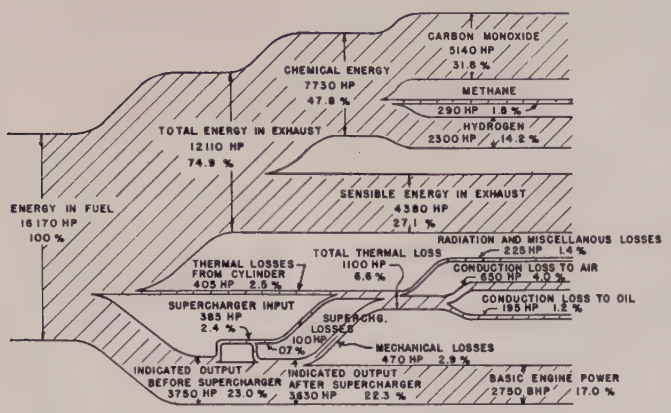


Fig. 12 – Turbo Compound engine heat balance – basic engine, take-off power, 2750 bhp 2900 rpm, 0.100 F/A ratio

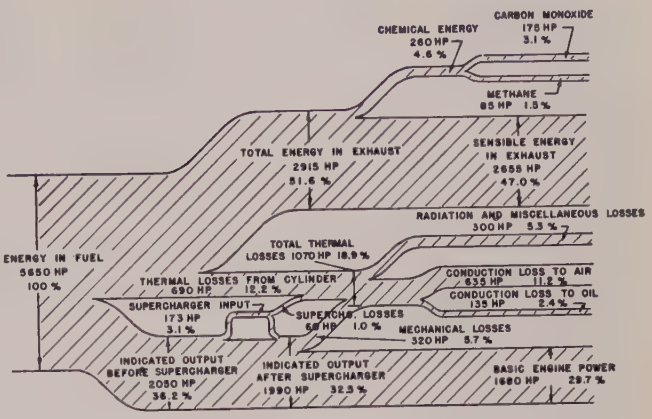


Fig. 13 – Turbo Compound engine heat balance – basic engine, cruise power, 1680 bhp, 2400 rpm. 0.056 F/A ratio

the blowdown process, engine back-pressure build-up is small. This is not the case when a pressure-type turbine is used. The resulting back pressure imposes a penalty on the basic engine by lowering the detonation-limited power of the engine and reducing engine critical altitude by reducing the engine airflow and horsepower at full throttle.

It is estimated that engine output at take-off using a suitable pressure turbine would be increased by approximately 8%, as compared to an 18% increase obtained with the present Turbo Compound using blowdown turbines.

At take-off power, thermal efficiency of the basic engine is such that only 17.0% of the energy available in the fuel is converted into brake horsepower, as shown in Fig. 12. 27% of the energy available in the fuel is rejected as sensible exhaust gas energy. At cruise power a greater percentage of the available energy – 30% – is converted by the engine into brake horsepower since combustion occurs at or near a fuel-air mixture conducive to best economy, as shown in Fig. 13. As a result of combustion with nearly chemically correct mixture, the sensible energy available in the exhaust gas is increased to 52%.

Of the sensible energy available in the exhaust gas at take-off power, 49% is in the form of kinetic energy available for power recovery, as shown in Fig. 14, the remaining portion being in the form of nonusable heat. Of the kinetic energy available approximately one-half is lost in the expansion process across the exhaust valve. This loss is a function of the exhaust valve opening rate and expansion from a small high-pressure area into a large low-pressure area. In the power recovery turbines the total aerodynamic, thermodynamic, and mechanical losses account for an additional 15%. The available kinetic energy for cruise operation is 35% of the sensible energy, as shown in Fig. 15, and is less than that for take-off operation since more energy is liberated in the form of nonusable heat. The loss in the power recovery turbines is reduced to 10%, due to the lower wheel speed and exhaust gas velocity. Although the net recovered power is a small percentage of the sensible energy, 43% of the energy available for power recovery is converted by the turbines into engine horsepower. That is, 500 hp at take-off power and 160 hp at cruise power account for an 18% and 10% increase in basic engine brake horsepower, respectively, at sea level. At critical altitude the recovered take-off power increases to 640 hp and at cruise power to 290 hp, which amounts to 26% and 19% increases in basic engine hp, respectively.

A cross-sectional view of the Turbo Compound engine is shown in Fig. 16. The power section consists of nine cylinders in each row bolted in a radial position to a steel crankcase. Power is de-

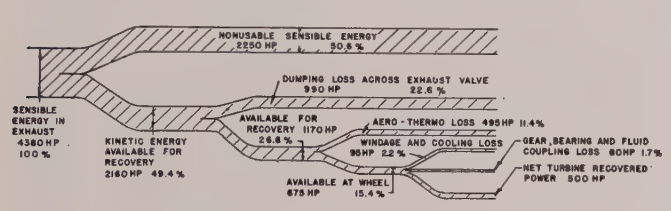

NONUSABLE SENSIBLE ENERGY
2250 HP 50.8 %
DUMPING LOSS ACROSS EXHAUST VALVE
990 HP 22.6 %
SENSIBLE ENERGY IN EXHAUST
4380 HP
100 %
AVAILABLE FOR RECOVERY 1170 HP
26.8 %
AERO-THERMO LOSS 495 HP 11.4 %
KINETIC ENERGY AVAILABLE FOR RECOVERY
2160 HP 49.4 %
WINDAGE AND COOLING LOSS
95 HP 2.2 %
GEAR, BEARING AND FLUID COUPLING LOSS 80 HP 1.7 %
NET TURBINE RECOVERED POWER 500 HP
AVAILABLE AT WHEEL
675 HP 15.4 %

Fig. 14 – Turbo Compound engine power recovery – takeoff power, 3250 hp, 2900 rpm, 0.100 F/A ratio, sea level

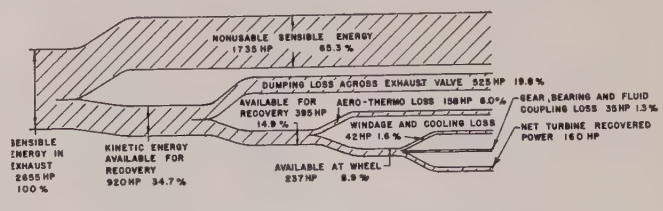

NONUSABLE SENSIBLE ENERGY
1735 HP 65.3 %
DUMPING LOSS ACROSS EXHAUST VALVE 525 HP 19.8 %
AVAILABLE FOR RECOVERY 395 HP
14.9 %
AERO-THERMO LOSS 150 HP 5.6 %
GEAR, BEARING AND FLUID COUPLING LOSS 35 HP 1.3 %
WINDAGE AND COOLING LOSS
42 HP 1.6 %
NET TURBINE RECOVERED POWER 160 HP
SENSIBLE ENERGY IN EXHAUST
2655 HP
100 %
KINETIC ENERGY AVAILABLE FOR RECOVERY
920 HP 34.7 %
AVAILABLE AT WHEEL
237 HP 8.9 %

Fig. 15 – Turbo Compound engine power recovery – cruise power, 1840 hp, 2400 rpm, 0.056 F/A ratio, sea level

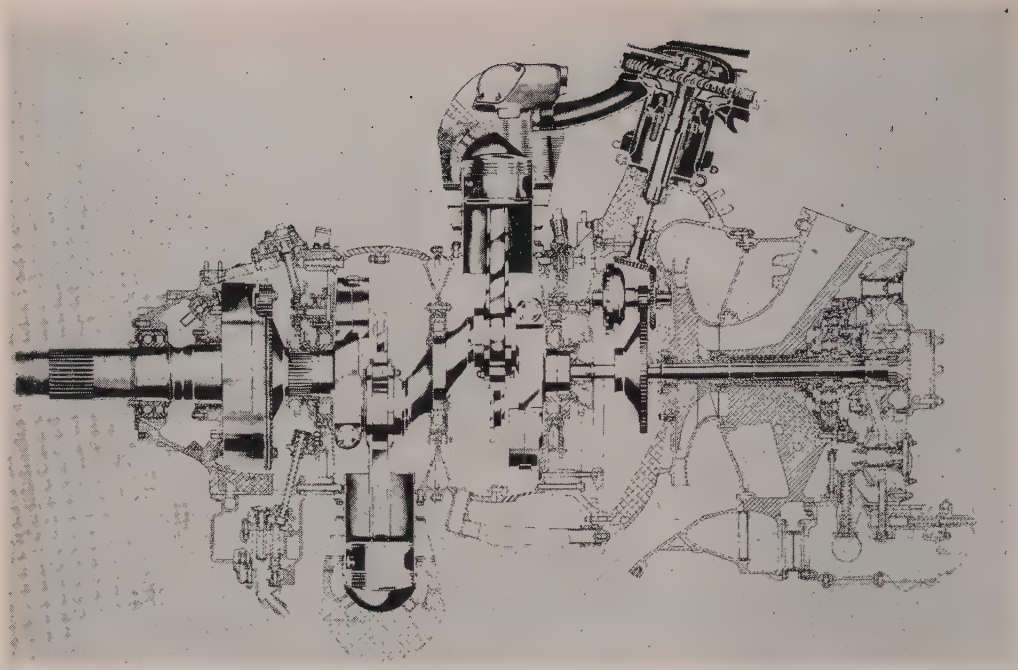


Fig. 16 – Cross-section view of Turbo Compound engine

livered through an articulated connecting-rod system to the crankshaft, which is supported in the crankcase by three roller bearings. The power recovery turbines are situated to the rear of the cylinders and are supported by the supercharger front housing at three equally spaced attaching pads.

The power recovered by the turbines is transferred into the rear of the crankshaft by means of a gear train operating in conjunction with three fluid couplings. The combined cylinder and turbine power is transmitted from the crankshaft to the propeller shaft through simple planetary reduction gearing located in the crankcase front section.

The engine incorporates a 2-speed gear-driven supercharger for altitude operation. The supercharger drive consists of two planetary gear trains driven by an extension shaft coupled to the rear of the engine crankshaft. The engine-driven accessories and ignition generators are installed on the engine rear section cover and are also driven by this same extension shaft through a suitable gear train. An engine-driven oil supply pump is located in the engine rear sump and supplies pressure oil to the entire engine. In addition to the main oil return pump, which is also situated in the rear sump, an auxiliary scavenge pump is provided in a sump attached to the crankcase front section to ensure adequate scavenging of the engine compartments.

The Turbo Compound cylinder, as shown in Fig. 17, was developed from the C18BD1 forged cylinder, which is now operating with good durability up to 10,000 hr in airline service. Improvements include generous radii at stress concentration points, more cooling fin area, and additional attaching threads. The cooling fin area has been increased about 40% on an overall basis, with local areas on the head receiving as much as a 200% increase in fin cooling area. These increases have resulted in a generally lower average temperature at the seat locations with significant improvement in performance. Naturally, these changes result in increased weight and the cylinders on the Turbo Compound engine weigh approximately 110 lb more than those on the C18BD engine.

Exhaust Valves – The Turbo Compound engine employs a domed-head, hollow-stem, sodium-cooled valve at both intake and exhaust locations, similar to those employed in earlier C18 engines. A valve installed in a cylinder is shown in Fig. 18. Development of these valves has included refinement of manufacturing methods to provide better internal finishes on the as-forged surfaces and improvement of the control of internal radii and surface finish in the internally machined area. The valve stem diameter has been increased to ¾ in. – as compared to 11/16 in. on earlier models – to provide an extra margin of strength. However, the throat restriction area remains unchanged to insure the proper heat balance and temperature distribution between the valve head and valve guide.

Numerous other valve materials and methods of fabrication have been tested experimentally. With the proper head coating, the Inconel M valve has equal resistance to pounding, radial cracking, and burning as compared to that of our standard nichrome-faced valve. However, stem wear is high with Ni Resist 1A, and Inconel has proved incompatible with our present HC-250 guide material.

Development testing of numerous new valve materials and combinations of these materials is in process. In addition to the development of these

materials applied to conventional valve designs, a considerable amount of testing is being conducted on valves completely machined internally as well as externally.

Exhaust Valve Guides – Development of the present exhaust valve guide (shown in Fig. 19) is the result of approximately 13,000 hr of single-cylinder testing. Some 80 combinations of stem and guide materials and configurations were investigated or tested. Early in the course of this investigation the mixture strength was found to have a profound effect on guide wear. Dezincification, a phenomenon of copper-based alloys recognized many years ago, was found to be the cause of excessive wear of bronze guides when lean mixtures (F/A below stoichiometric) were used. Wear at rich mixtures was low. However, with most cast irons, including many high-alloy irons such as Ni Resist 1A, the reverse wear pattern was true. Short periods of rich-mixture operation caused excessive wear with Ni Resist 1A guides.

The present HC-250 guide material is a high-carbon (2¾%) high-chromium (25%) cast structure bordering on tool steel analysis. Little wear of either the guide or valve stem was encountered regardless of mixture strengths. During the development of this material, it was found to be somewhat incompatible with soft stem valves, in-

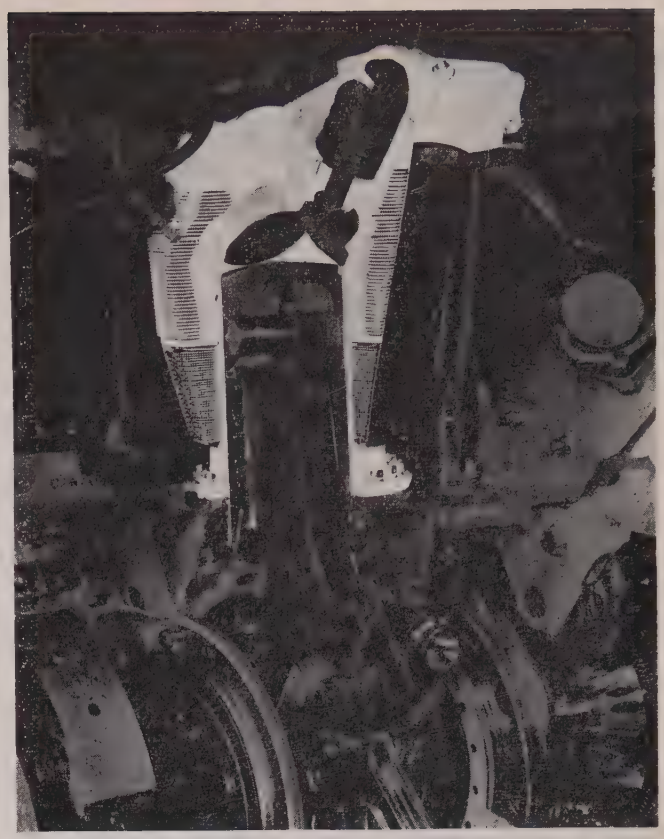


Fig. 17 – Cylinder accentuated. This cylinder was developed from C18BD1 forged cylinder

Fig. 18 – Exhaust valve

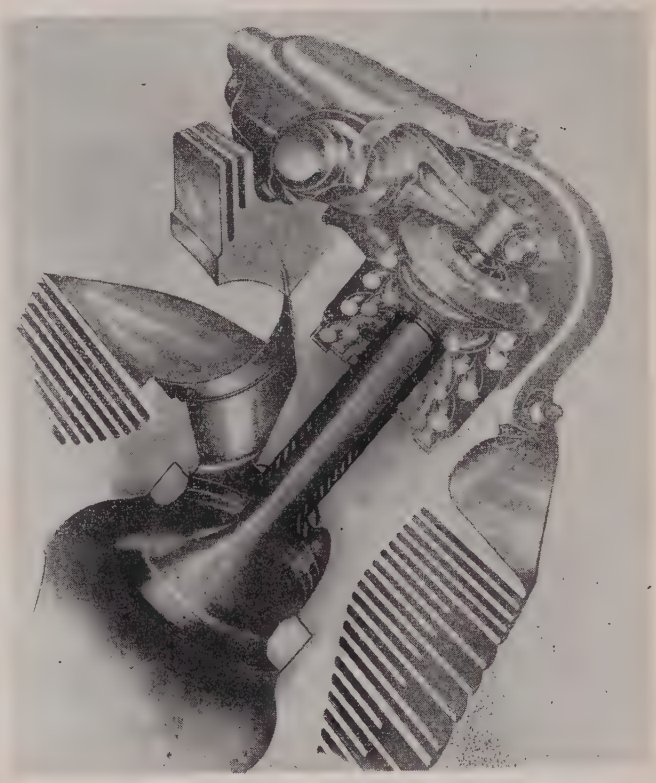


Fig. 19 – Exhaust valve guide

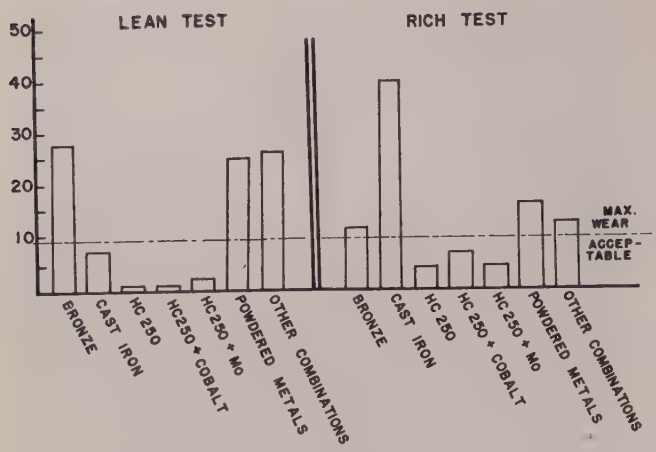


Fig. 20 – Summary of relative exhaust valve guide wear – standard valves

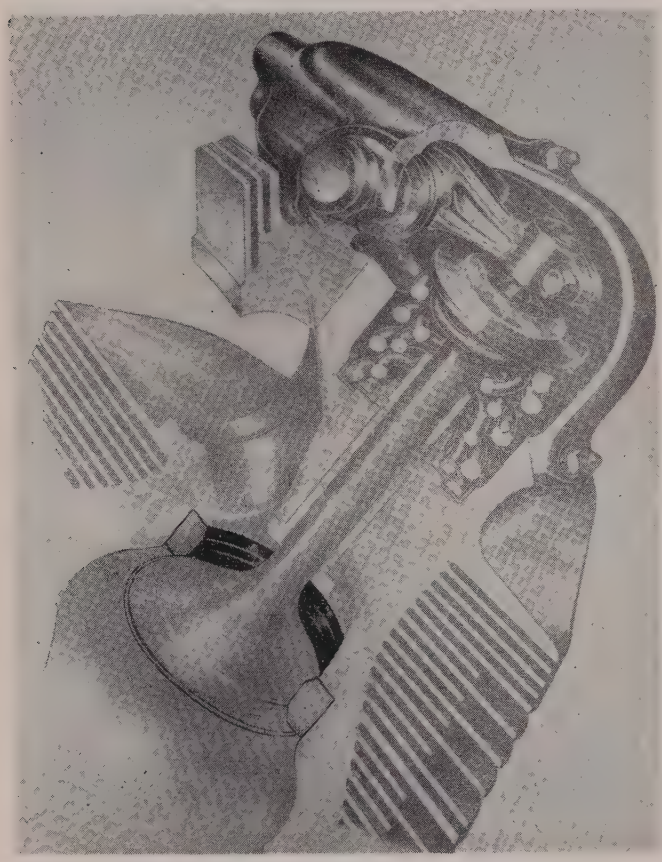


Fig. 21 – Exhaust valve seat

cluding shotpeened Inconel M. Colmonoy No. 6 coated stems were fairly successful on experimental test and service testing is proceeding to evaluate this combination. Experimentally, other stem treatments are being tested in conjunction with HC-250 guides and other guide materials are being investigated for resistance to wear during both rich and lean operation. A summary of experimental testing of guides is shown in Fig. 20.

Valve Seats – Fig. 21 shows a valve seat installed

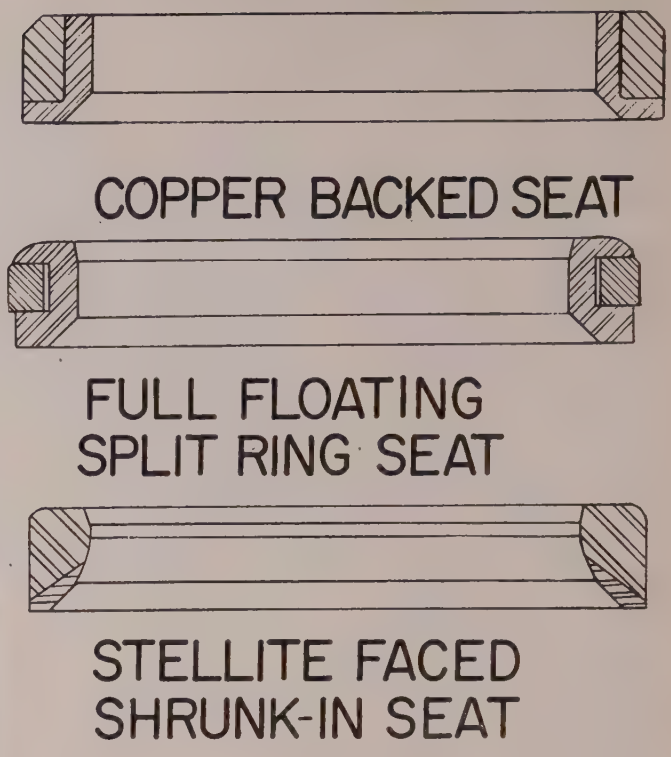


Fig. 22 – Valve seats

in a cylinder. Valve-face and valve-seat-face burning has long been a major combustion-chamber problem for most engine manufacturers. As has been pointed out in several recent papers, valve rotation helps to minimize the problem. However, rotation is not a basic correction. Where cylinder-head cooling is sufficiently uniform the seat need provide nothing more than a hard, burn-and-wear-resistant surface to effect a good seal. However, because of space and weight limitations, a compromise must be made from optimum cooling to obtain a workable configuration. Hence, the floating-type seat (shown in Fig. 22) was developed to minimize seat distortion. The earliest production floating seat was the buttress thread seat of the Cyclone 9-cyl military engine. This has been developed to the more durable stellite split-ring seat. Although operating at a higher overall temperature, this seat has quite uniform temperature around its circumference due to the partial insulation of the seat from the head by the double interface between the seat-to-ring and the ring-to-head.

This split ring provides the added advantage that ovalizing forces from uneven temperatures around the valve-seat counterbore are not imposed upon the seat. Consequently, it can remain round and provide a seal with the valve face.

Another approach to the maintenance of circularity of the valve seat is that of providing a good heat conducting material in the seat itself. In addition to reducing the temperature differential around the seat, this configuration transfers heat

Fig. 23 – Crankshaft accentuated

more effectively from the valve to the cylinder head. Considerable experimental test experience has indicated that a copper-backed seat is a close second to the split-ring type of seat with respect to seat burning. In addition, this type of valve seat is considerably less expensive to manufacture.

Main Crankcase and Crankshaft – The crankcase main section and crankshaft assembly is shown in Fig. 23. A 4-piece crankcase main section fabricated from steel forgings with provisions for a crankshaft main bearing in each diaphragm was selected for this model. This crankcase arrangement had proved itself in earlier models of the R-3350 Cyclone series.

The crankshaft is a conventional 180-deg opposed 2-throw shaft composed of three sections. The front and rear sections each have a supporting roller bearing mounted outboard of the crankcheek blade and a friction clamp to secure them to the center section. The center section is formed by the two crankpins separated by a large roller bearing.

Since the developed shaft horsepower in this model has been increased by improvement to the basic engine and the addition of the three exhaust turbines, it was necessary to increase the strength of the crankshaft assembly. Preliminary testing indicated that maximum stresses were induced in the crankshaft by engine-induced vibrations with the natural frequencies found at or near take-off rpm. To raise the natural frequency of the crankshaft, material was added to the center crankcheek. In addition, the rear crankcheek thickness was increased to strengthen and stiffen this member of the assembly.

Supercharger Drive – A 2-speed supercharger drive similar in design to that provided in earlier models has been provided for this engine. The low ratio drive is accomplished through a primary planetary drive with a fixed outer ring gear and a secondary planetary drive with all components free to rotate. In low ratio the drives are coupled by a roller clutch. High ratio is obtained by stopping the pinion carrier in the secondary drive with an oil-operated multiplate friction clutch permitting the roller clutch mounted on the carrier to override.

Two versions of this type of drive have been developed for the Turbo Compound engine, as shown in Fig. 24. The first type incorporates pinions with integral journals, while the second type incorporates trunnion-mounted pinions. The latter configuration was chosen for the current models

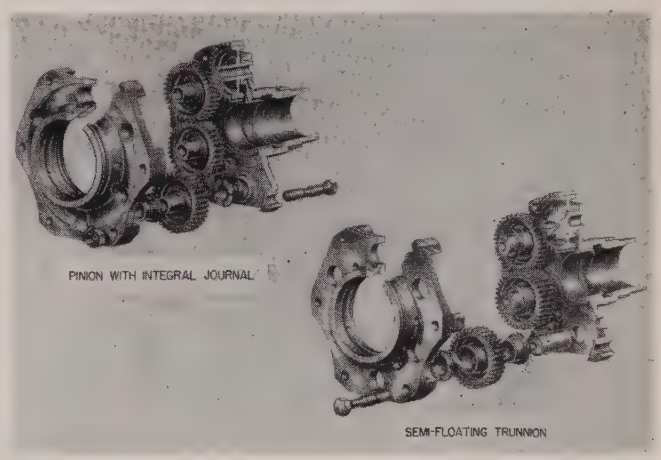


Fig. 24 – Secondary impeller drive pinion and carrier assembly

because of reduced unit bearing loads and reduced cost of replacement parts at overhaul.

A disc-type main engine pressure oil filter (shown in Fig. 25) has been incorporated to prevent foreign material contamination of the bearing surfaces.

Development of Turbine

A sectioned drawing of a power recovery turbine is shown in Fig. 26. The turbine is comprised of four basic systems – the power system consisting of the nozzle, wheel, and shaft; the cooling system, including the cooling shields and impeller; the vibration damping system; and the lubrication system. The parts comprising these systems are supported by two basic structural members. The first, the nozzle support sustains the stationary parts including the aircraft supplied exhaust shroud, while the second, the shaft support, sustains the rotating parts. Both the nozzle support and shaft support are clamped and indexed to an adapter bolted to the engine front supercharger housing.

Lubricating oil is supplied to the turbine unit from a drilled passage in the front supercharger housing. The oil is introduced to the turbine by an annulus at the lower end of the shaft support.

Sealing is accomplished by O-rings installed in grooves on either side of the oil annulus. Oil flows across the lower bearing, through the void between the support and shaft, and then across the upper bearing. A seal from the "hot" section of the turbine is effected by a Graphitar seal ring and a stationary bellows seal. Oil scavenging is accomplished by gravity drain to the interior of the front supercharger housing.

The cooling system components of the turbine are emphasized in Fig. 27. Ram air, ducted from the front of the engine, is introduced into the turbine through an enclosing muff. An impeller, installed on the main shaft, provides the necessary pressure rise to force the air through holes at the base of each wheel bucket. The air is collected on the outlet side by a shield assembly and discharged into the exhaust outlet. Sealing between the exhaust gas and the cooling air is accomplished by a concentric groove labyrinth seal on the underside of the cooling air impeller and by a step-type seal on the upper side of the wheel.

Recently, marginal turbine wheel hub cooling was encountered during high-power operation at

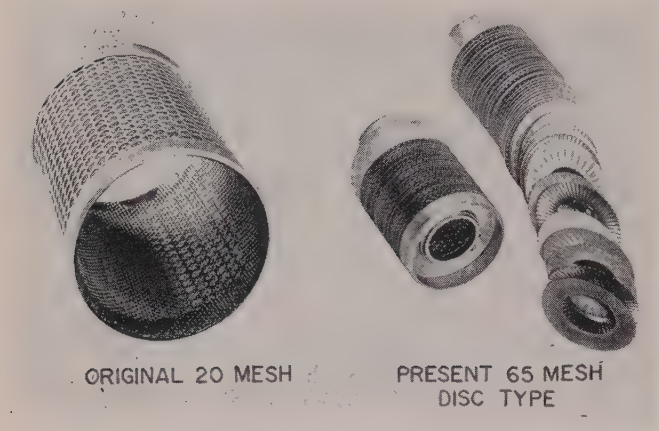


Fig. 25 – Oil inlet strainer

Fig. 26 – Cross-section of turbine

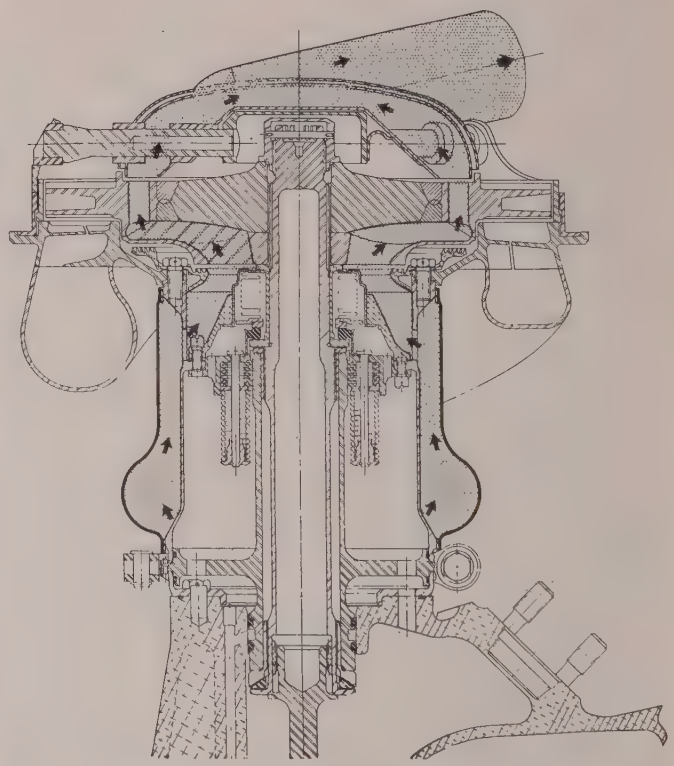


Fig. 27 – Cross-section of turbine with cooling air path indicated

high altitude. In the particular installation, it was found that the pressure differential between the exhaust system and the cooling system increased at high altitude and disturbed the flow of air above the wheel. While this condition was improved somewhat by modifications to the flight hood, other changes as shown in Fig. 28 were made to the cooling air system to increase the flow in the cooling system. A tangential outlet was provided to minimize the back pressure on the cooling impeller. To reduce exhaust gas leakage into the cooling air system a vent arrangement was incorporated at the step-type seal on the upper side of the wheel and this exhaust gas was drawn off in an outlet concentric with the cooling air outlet to avoid dilution of the cooling air above the wheel. An inner heat shield was also provided to reduce the effect of radiant heat on the wheel hub. As shown in Fig. 28 this modification resulted in 100% improvement in cooling at 18,000-ft altitude and permitted satisfactory operation under any power condition in the airplane.

The turbine shaft is sustained and aligned by two steel-backed silver bearings mounted at either end of the shaft support, as shown in Fig. 29. Axial thrust is absorbed through a steel-backed silver spherical thrust washer at the lower end.

The shaft material is stainless steel for corrosion resistance and strength at the elevated operating temperature. The shaft bearing journals are nitrided for wear resistance and flats are provided on each journal to assure adequate oil flow at high speeds.

End float of the shaft is maintained within specific limits by selection of the proper offset thrust washer seating on the end of the upper journal. Radial grooves are provided across the thrust flange of the upper bearing to assure adequate oil flow under minimum end float conditions.

The cyclic power impulses inherent in a blowdown system tend to induce a whirling vibration in the rotating components. The vibratory forces at the resonant points, if undamped, would be transmitted to the turbine and turbine support structure, necessitating a very heavy, rigid construction. Since there are a number of exciting frequencies generated by both the engine and the turbine gas forces, it was necessary to develop a damping system that would prove effective for all orders and have a reasonable weight. This was done by designing the turbine support structure as a spring, which is a pedestal on a spring diaphragm permitting the turbine wheel within the limitations of the spring, to move slightly in all radial directions. The upper end of the pedestal was equipped with a plate-type friction damper in which half the plates were fixed to the nozzle support and half the plates were fixed to the pedestal. The vibration damper system is accentuated in Fig. 30. Loading was provided between the plates by eight springs compressed to a predetermined load. Since relative motion is allowed by this means and damping is present, vibratory gas forces coming through the wheel or engine-excited forces transmitted through the case cannot excite resonant vibration over the entire operating range of the engine.

The materials incorporated in the turbine wheel (shown in Fig. 31) were selected for optimum corrosion resistance and strength at elevated temperatures. The buckets are Haynes Stellite No. 31, which not only has excellent corrosion resistance, but also a low creep rate at the operating stress

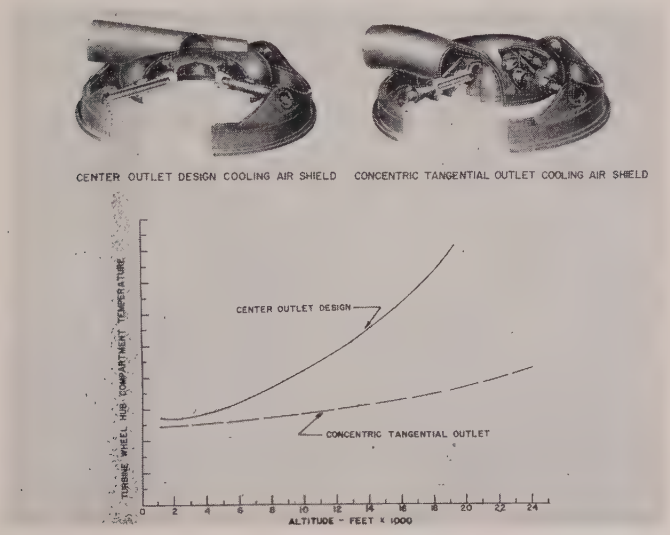

CENTER OUTLET DESIGN COOLING AIR SHIELD
CONCENTRIC TANGENTIAL OUTLET COOLING AIR SHIELD
TURBINE WHEEL HUB COMPARTMENT TEMPERATURE
CENTER OUTLET DESIGN
CONCENTRIC TANGENTIAL OUTLET
0 2 4 6 8 10 12 14 16 18 20 22 24
ALTITUDE - FEET X 1000

Fig. 28 – Power recovery turbine cooling air shield development, with temperature curve

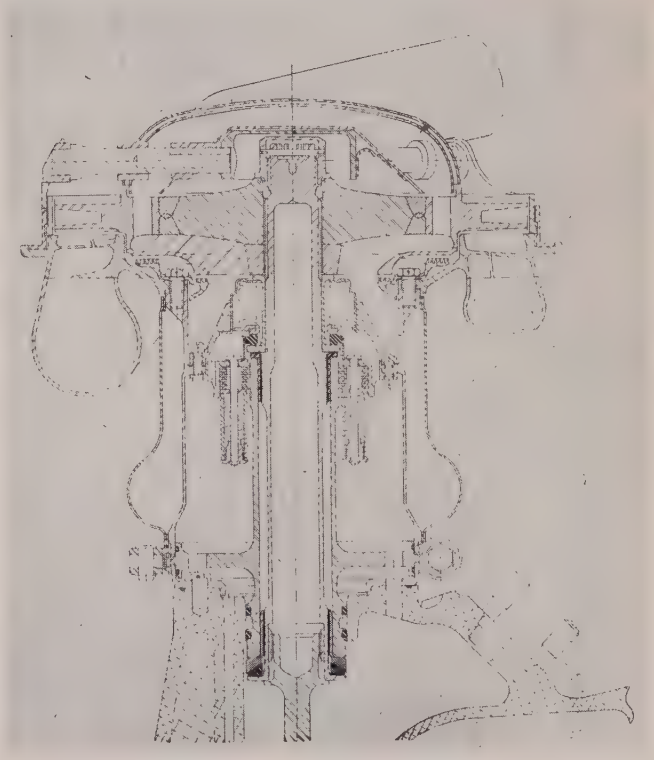


Fig. 29 – Cross-section of power recovery turbine with bearings accentuated

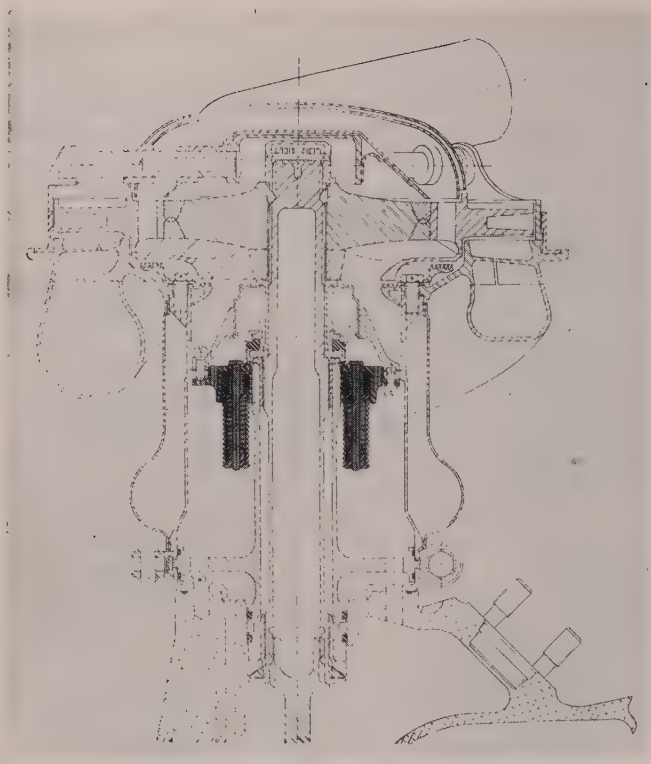


Fig. 30 – Cross-section of power recovery turbine with vibration damper accentuated

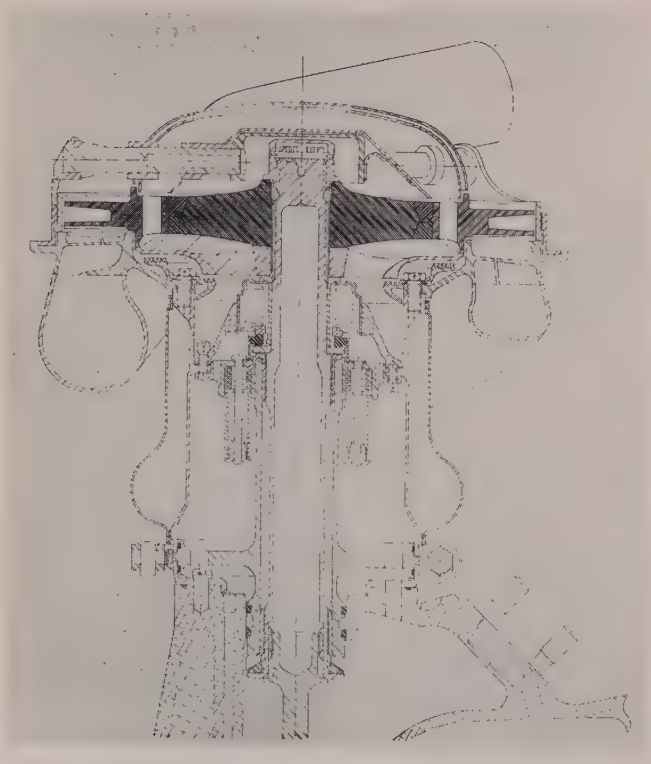


Fig. 31 – Cross-section of power recovery turbine with wheel accentuated

and temperature level. The buckets are welded to the rotor disc since this method of attachment provided the lightest design.

Inconel X was initially selected for the disc material because of its high-temperature properties. Testing has shown, however, that the operating temperature level of the disc is such that less strategic alloys may be employed. Evaluation of Timken 17-22A material with an oxidation-resistant coating is currently in progress.

The wheel assembly is splined to the turbine shaft with a sliding fit when cold. Primary torque transmission, however, is accomplished through a friction joint, since the wheel is clamped through a spool spacer against the end of the upper shaft journal. Clamping magnitude is determined by measuring and controlling shaft stretch within predetermined limits.

To isolate the torsional vibrations inherent in the power recovery system from those inherent in the basic engine, a fluid coupling (shown in Fig. 32) is provided in the drive system. This coupling prevents any buildup of forces in the drive system excited by torsional vibration and permits the use of a lightweight drive system. To avoid complications in the lubricating system, engine oil is used in this coupling. Early in the service operations it was evident that, in spite of the high-velocity circulation of oil within the fluid coupling, the quantity of oil flow through the coupling was insufficient to flush completely the low-velocity discharge area where sludge would accumulate. This locked the two elements of the fluid coupling, which permitted torsional excitation of the drive and adversely affected its durability. Increasing the oil flow and improving the cleaning of the airplane oil system improved the service life but did not completely eliminate the accumulation of the sludge. As further insurance against sludge accumulation, several configurations were developed that, by mechanical means, prevented formation of sludge in the low-velocity area. The most desirable configuration, however, was the vortex-flushed fluid coupling, which is currently incorporated in the engine. This coupling utilizes the high-velocity stream of oil circulating within the coupling to flush the area of potential sludge accumulation. The original and the vortex-flushed fluid coupling are shown in Fig. 33.

Development of Exhaust System

Early in the design study of the exhaust system for the Turbo Compound engine it was recognized that the ultimate configuration would have to be based on test experience rather than trying to anticipate all problems in advance. Consequently, the initial configuration was simple and light. Siamesing was adopted to reduce the system complexity; plain sliding joints were provided; the pipe metal thickness was established at the most practicable minimum. Modifications were made

Fig. 32 – Cross-section of engine with fluid coupling accentuated

before arriving at the final configuration, shown in Fig. 34, but the basic concepts are still present.

During the initial development of the exhaust system, the exhaust pipes were fabricated from seamless tubing since changes in contour and/or configuration could be accomplished rapidly and easily. These pipes were not ideal from a durability standpoint, possessing inherent weakness at welds joining siamesed sections. Once the configuration was established a change to the half-stamped method of fabrication was initiated on a production basis. A comparison of the two types of fabrication is shown in Fig. 35.

Initially, the exhaust system between the cylinders and the turbines incorporated simple slip joints to permit thermal expansion of the pipes

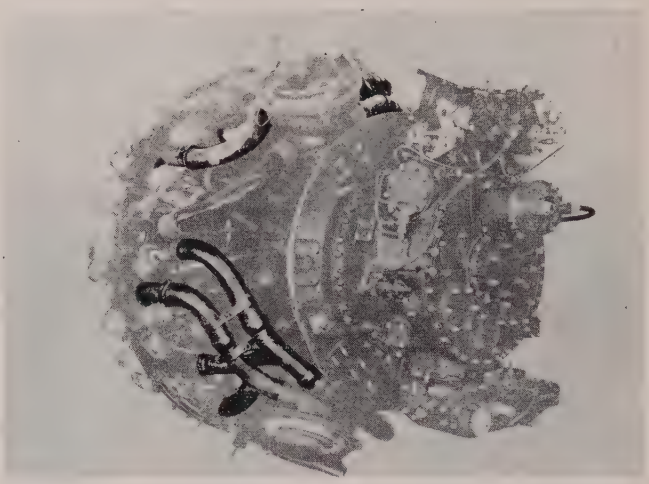


Fig. 34 – Rear quarter view of engine with turbine inlet pipes accentuated

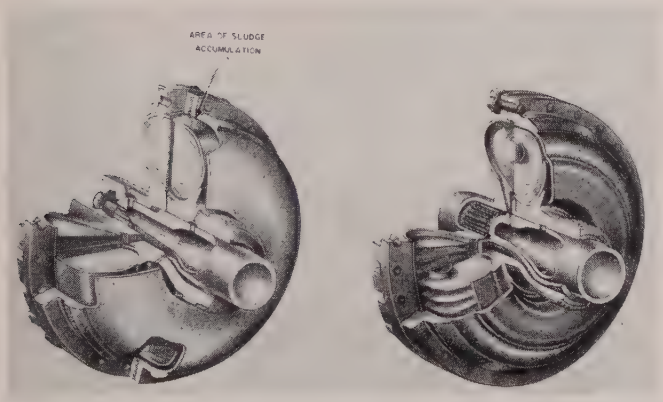


Fig. 33 – Power recovery turbine drive fluid coupling – left: original design; right: present vortex – flushed design

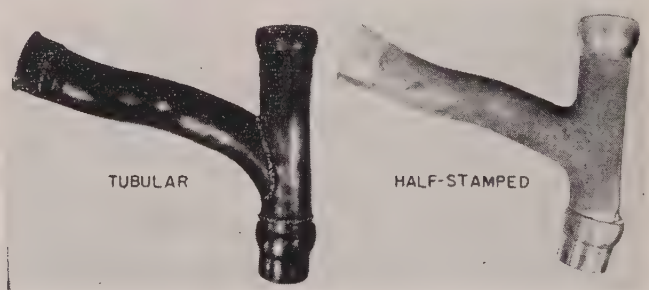


Fig. 35 – Turbine inlet pipe

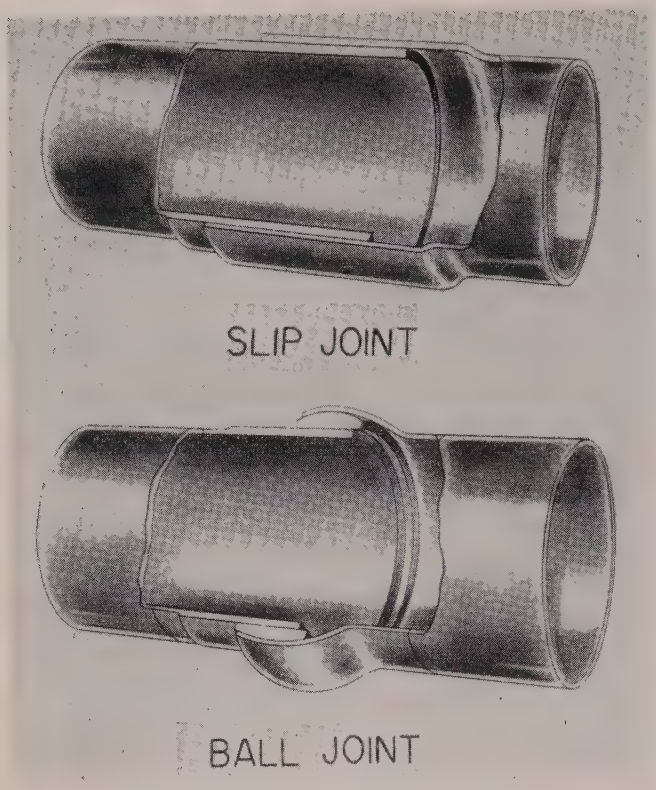


Fig. 36 – Turbine inlet pipe expansion joint

Fig. 37 – Figure-8 clamps shown installed on exhaust pipe

during operation, as shown in Fig. 36. This provided a relatively rigid system of fair durability. The effect of the system was such, however, that thermal and gas loads induced high stresses at the nozzle inlet joint and considerable difficulty was experienced with cracking of the sheet-metal inlet portion of the nozzle. To reduce this type of loading, flexible ball slip joints were incorporated in the pipe connections and at the nozzle inlet joints, as shown in Fig. 36.

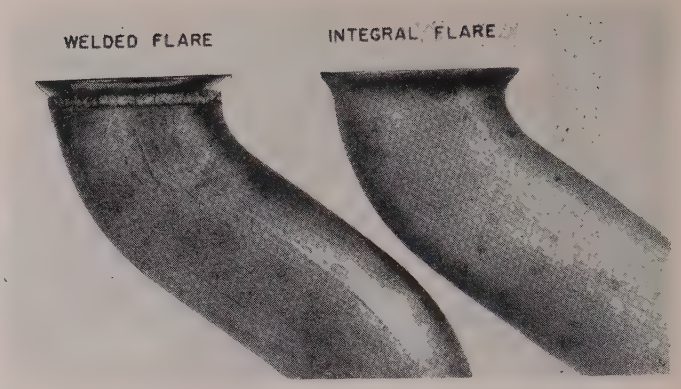


Fig. 38 – Cylinder end of front cylinder exhaust pipe

Design of the original ball slip joints was based on the current practice in use on existing exhaust manifold systems. Testing of these modifications indicated excessive wear and pounding. To determine the cause of this difficulty, temperature measurements of the slip ball joint components were taken during engine operation. These measurements indicated the temperature differential between components of the joint was considerably less than originally assumed. Consequently, the clearances in these joints were reduced, resulting in a marked improvement in the durability of the flexible joints. Coincident with these modifications, the axial bearing length in the joint was increased to reduce the unit loading. To provide stability of the exhaust pipes in the vicinity of the flexible joints, figure-eight-type braces (shown in Fig. 37) were installed between adjacent pipes. These braces, in addition to stabilizing the pipes, also maintain the alignment of the system at the joints.

Development of the exhaust system was accomplished with pipes incorporating a wall thickness of 0.047 in. This relatively thin wall was employed to provide as light an exhaust system as possible, consistent with satisfactory durability. After this system had proved its durability, the wall thickness of the pipes was increased to 0.062 in. to assure better long-term wear characteristics.

Originally, the cylinder end of the front cylinder exhaust extension incorporated an integral flare, which mated with the exhaust port connector. Early testing resulted in cracking in the flare, and difficulty in maintaining a uniform thickness in the sheet metal due to the relatively deep draw. To eliminate this difficulty a machined flare, butt-welded to the end of the pipe, was incorporated, as shown in Fig. 38. Difficulty with cracking adjacent to the weld was encountered with this design. The integrally flared pipe was redesigned to provide generous radii between the pipe diameter and flared portion, which resulted in a uniform wall thickness, and "coining" of the flare interior was incorporated

to assure a good fit on the cylinder connector, as shown in Fig. 38.

Besides the function implied by its part name, the turbine wheel hub cooling shield must prevent mixing of the cooling air with the exhaust gas and provide a shroud ring for the turbine wheel. These functions must be maintained over a wide temperature range, which presented a number of problems determinable only by actual experience. The sequence of development is shown in Fig. 39.

The initial configuration incorporated slip joints at the junction of the supporting bolts, outer flange, and sheet-metal bushings to permit thermal expansion. Operation showed severe pounding at both locations.

The first approach toward correction of the trouble was to increase the bearing area by providing 2-piece supports with a long engagement length. This design gave trouble in that the two sliding members tended to seize in the extended position during operation. In many cases failure of the lugs occurred from the resulting overstress.

At this point the method of approach was altered toward increased strength and rigidity. The current configuration consists of a completely welded unit with additional braces between the shield assembly and inner flange, and wider lugs on the outer flange support.

There have been, of course, other problems associated with the development of the Turbo Compound engine that have not been discussed. These problems were similar to those normally encountered during the development of any reciprocating engine. The items presented were selected on the basis of their importance and engineering interest.

Future Possibilities

The present Turbo Compound engine has recently entered the commercial field and, in its present state, is certain to remain there for some time as well as in military patrol and transport action. Use of this engine in forthcoming installations will largely place it in competition with turboprop or turbojet units of higher equivalent ratings and better power/weight ratios. In spite of these factors, development of present designs continues to make the Turbo Compound engine attractive for future applications up to a speed of about 450 mph.

Future prophets predicting the demise of the reciprocating engine should take note that past prophets have not been too successful. When one considers the overall costs, fuel consumption, high initial cost, and the inherent operating problems of turbine-powered aircraft, it is difficult to conceive that the piston-engine transport will become an extinct species within the next 10 years. Consequently, our development of the Turbo Compound engine is proceeding industriously and will result in considerable increases in both cruise and take-off horsepower. Application of these advanced engines to transport airplanes will continue to provide fast, efficient, and profitable transportation for many years to come.

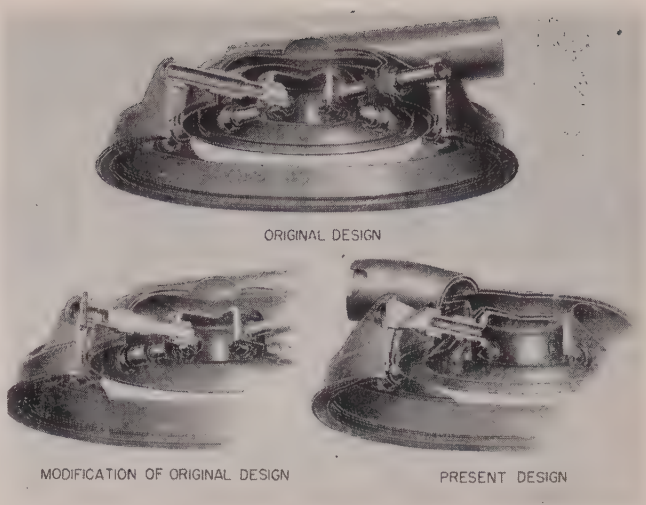


Fig. 39 – Cooling shield

ORAL DISCUSSION

Reported by R. F. Schwarzwalder
Wright Aeronautical Division, Curtiss-Wright Corp.

S. M. Udale of Holley Carburetor asked a question pertaining to WAD experience with overrich mixture and if so what were the results.

Mr. Eichberg said that the exhaust gas temperatures approximated 1430-1450 F at the turbine cooling cap during take-off, and that at cruise speed it approximated 1550 F at the turbine cooling cap. He pointed out that in one military installation, the P2V Neptune patrol bomber, there was a problem during the descent with the engine speed at 2600 rpm and low power with rich mixture, which resulted in afterburning. In an effort to overcome this along with a temperature of approximately 1850 F, the inner shield was removed from the cooling cap. The fuel-injection engine in this regard gives much better results than one not having this feature, due to leaner mixtures and reduced turbine wheel speed.

H. J. Buttner of Continental Aviation, asked whether the same or similar problems exist on the intake valve and guide as on the exhaust valve and guide, particularly regarding the valve spin. He further asked if efforts were being made to improve lubrication and load on the intake valve guide. He stated that he assumed no problem existed but was asking these questions because statements were made on oxidation of intake guide materials.

Mr. Eichberg replied that the intake valve guide is made of bronze and no problem exists as on the exhaust side of the cylinder head. This material has been in use since 1942.

He also pointed out that both intake and exhaust rocker boxes are pressure fed with gravity drain. However, on the intake side, he said, there is no oil spray on the valve stem itself but that there is a spray into the whole box.

Regarding valve materials a problem arose because of spalling on the stem. WAD tried coatings of various types and the best results were obtained with Calmonoy. Service tests with HC 250 material used on the guides on Constellation engines shows up well at the 1200-hr overhaul.

Mr. Eichberg mentioned further that valve stems made of Inconel wore more when used in conjunction with HC 250 material guides.

Measurement and Control of Residual Stresses

CONTROL of residual stresses in engineering materials has received increasing attention in recent years. Designers have become aware that the strength and life of many parts are affected to a major degree by factors other than the computed stresses resulting from loads applied in service. Notch effects, surface finish variables, nonuniform thermal environments, and residual stresses are only a few of the factors that are difficult to include with precision in stress calculations. In particular, the residual stresses present as the result of the production history of the part may be extremely important. For some parts the significant history will include the manufacture of the raw material through diverse processing and fabricating operations and including assembly techniques.

Fig. 1 – General view of cylinder block in foundry

In general, it is the hope of designers that residual compressive stresses will be present on surfaces loaded in tension. To this end, mechanical working of such surfaces by shotpeening, rolling, or other methods is often used. These are very effective when methods are carefully selected and controlled. Compressive residual stresses can also be produced by nitriding. Carburizing and other thermal treatments are sometimes, but not always, favorable.

With the exception of the items noted, the designer ordinarily does not have control over residual stresses. It is important that production people realize the importance of manufacturing methods and process control on many types of such stresses. For example, it is known that the surface of cold-drawn steel bars may be either in tension or compression, depending on processing technique. Welded parts are a well-known example of complex residual stresses, not always favorable, resulting from processing methods. Castings in general are often subject to complex residual stresses, again not always favorable. We believe that designers should be familiar with such production problems that limit application of materials and methods. We also believe that with such familiarity the designer can frequently assist in reducing the magnitude of the problems, or help in their solution.

The purpose of this paper is to present information on some of the residual stresses found in automotive cylinder blocks. The methods used for quantitative measurement will be given, and the control techniques found useful will be shown. This particular work deals mainly with the straight-eight type of block, but it is believed the principles of measurement and control will be useful for a fairly broad field.

Control of residual stresses in cylinder blocks started at Packard about 15 years ago. At that

in Cylinder-Block Castings

R. E. VanDeventer and Forest McFarland,
Packard Motor Car Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1954.

time a number of castings were scrapped for cooling cracks in the valve compartment wall. Fig. 1 shows a general view of this type of block as it appears in the foundry. Most of the valve housing cracks occurred between the center cylinders. Some were observed in the foundry, others developed later in the machine shop, and still others occurred in service. In practically all cases the cracks were formed after the castings were cold. The total represented only a small percentage of production, making diagnosis quite difficult. Design changes were tried, such as a wavy valve compartment wall instead of straight, increased wall thickness, and the like, but without appreciable effect. Metallographic examination of cracked sections showed the coarse dendritic pattern typical of very slow cooling, and the Brinell hardness of these sections was quite low. Neither condition was noted on other portions of the same castings.

As a part of the investigation of this problem, cooling curve data were obtained at different points of the casting, with thermocouples located as in-

THE control of residual stresses in automotive cylinder blocks is discussed in this paper. According to the authors, this study was started because a small percentage of castings was being scrapped because of cooling cracks in the valve compartment wall.

They discovered that the greatest single factor in inducing trapped stresses in castings is too great a difference in the cooling rates for different portions between 1400 F and 600 F.

They also found out that residual tensile stresses can be eliminated in a particular section by keeping its cooling rate at least as fast as that of other significant sections. Moreover, it is possible to move trapped stresses from one section to another by overcooling.

They have also developed a method that gives quantitative results that serve as good index values of the trapped stresses present even in complex castings.

The Authors

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FOREST McFARLAND (M '28) is chief engineer, Design and Research Division, Packard Motor Car Co. He attended Michigan State College and received a B.S. degree in mechanical engineering in 1921. Mr. McFarland has been with Packard since 1924.

dicated. The center portion of the valve compartment wall, a point on the top face, and one of the end sections were used for comparison. Fig. 2 shows the curves at these points for the first 30 min after pouring. It will be noted that after the first 10 min the valve compartment wall is definitely the hottest of these three locations. Fig. 3 shows a continuation of these cooling curves for a block allowed to cool in the mold. The same relative difference continued over the entire period of cooling.

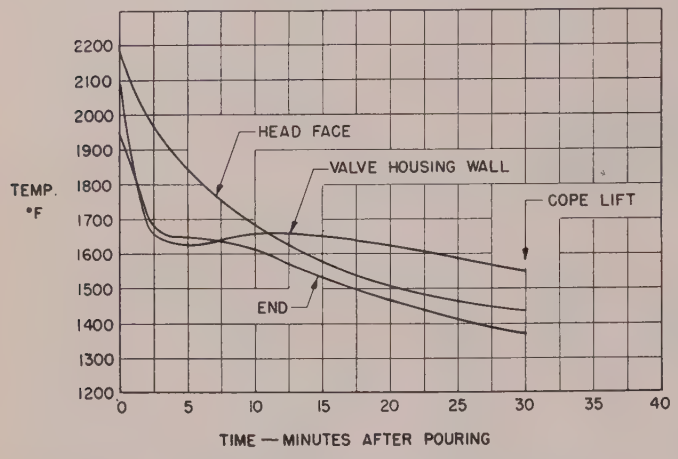

Fig. 2 – Cylinder-block cooling data – pouring to cope lift

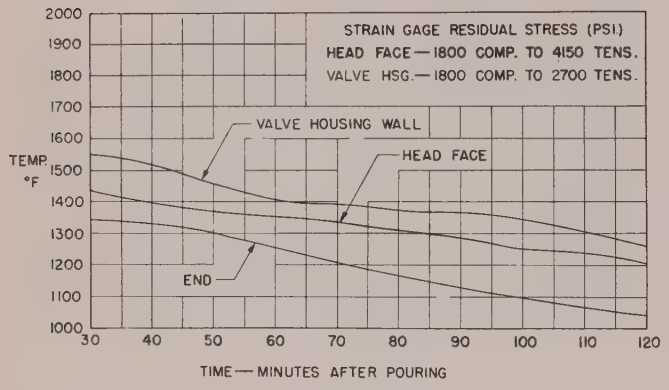

Fig. 3 – Cylinder-block cooling data – after cope lift – casting cooled in mold

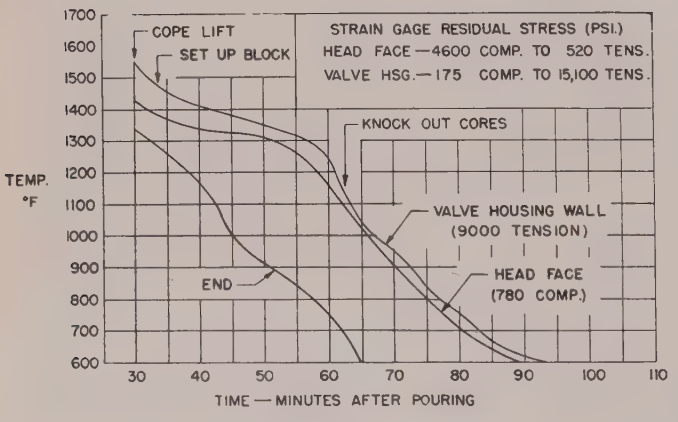

Fig. 4 – Cylinder-block cooling data – after cope lift – 1935 practice

It is obviously impractical to cool all castings in the mold, and Fig. 4 shows the cooling curves on blocks made using the practices in effect up to about 1935, which consisted in lifting the cope off after approximately 30 min and standing the casting upright. No particular effort was made to remove molding sand or cores from any specific surfaces. The curves show that in this case the valve compartment wall continued to be the hottest part of the casting, the cooling rate being considerably greater than when cooled in the mold. The top face cooled slightly faster than the valve compartment wall, but the ends of the casting cooled much more rapidly, due to better air circulation.

It was reasoned at that time that with these cooling conditions the external sections contracted more rapidly than the valve compartment wall, which was insulated by the heavy core, so that the total length was, in effect, controlled by these more rapidly cooled sections. This caused the valve compartment wall to yield while hot, due to plastic deformation, and when it tried to contract later it was restrained, either causing cracks or leaving trapped tensile stresses, which might be of any degree up to the strength of the material. This is similar to the familiar contracting stresses on welded parts held in a restrained position.

Action taken at that time included the standardization of inoculation by ladle additions of silicon alloys to obtain more uniform properties with the different cooling rates, and the addition of an operation to knock out the valve housing core immediately after the cope was lifted and the casting stood upright. This helped the valve housing wall to cool a little faster, but was not sufficient to eliminate the cracks completely. A compressed-air pipe was then installed in the cooling tunnel from the point where the core was knocked out up to the time of shakeout. This is illustrated in Fig. 5. A row of holes in this pipe directed air at the valve compartment to accelerate further the cooling of this part of the casting. Cracks then developed on the opposite side of the block, so a similar perforated pipe was installed on this side also. The total effect of these changes was to stop the cracks completely.

The effects of these changes in practice are given in the cooling curves of Fig. 6. This graph shows that the valve housing core and the molding sand near the distributor pad on the opposite side are broken with a spud soon after the casting is raised to an upright position. The broken valve housing core is then blown out, followed by cooling on both sides for about 8 min, as described. It will be noted that the valve housing wall now cools more rapidly than the head face from 1300 F down to 600 F. There is also substantially less temperature spread between the spring housing and the end section.

Over the succeeding years a number of different

engine designs and modifications were produced but, since the general configurations were similar, the same foundry practices were continued. Valve housing cracks over these years were, for all practical purposes, nonexistent. In 1950 our proving ground started reporting cracks on the top face of a few cylinder blocks after long mileage endurance testing. Comparison of designs with earlier models showed that we now had a thinner top deck for better cylinder and valve cooling, and the engines were producing substantially more horsepower than earlier models, but the changes did not appear to be sufficient to account for the difficulty. In 1951, with still more horsepower, reports of a few cases of similar cracks started appearing in service. In most instances the cracks occurred on the top face of the block between one of the center cylinders and its intake-valve port. Generally, the cracks had the pattern of fatigue failures, with the origin at the sharp corner at the top of the cylinder bore and progressing until a water leak developed.

The evidence at this point was not sufficiently complete to establish either foundry practice or engineering factors as the definite cause of trouble. Therefore, the engineering research and metallurgical departments teamed up to conduct a joint investigation. The engineering phase will be covered later. At this point it should be mentioned that the development of the method (which will be described later) for quantitative measurement of residual stress permitted an evaluation of the effects of variations in foundry technique. There are, of course, many variables in the production of complex castings that can affect the final stress condition. The technical literature gives a few references showing that in simple castings the stress can be changed with moisture content of the molding sand, with changes in pouring temperature, and possibly with mold or core strength. A definite effect is reported with change of time in the mold before stripping. While we recognize that these and many other items can affect the residual stresses, our work indicates that most are of a minor nature compared to the major item. We are satisfied that the relative cooling rates of different portions of complex castings after solidification are the most important. Our work, therefore, was concentrated on cooling rate analysis and control, and in particular that portion below 1400 F.

In order better to understand the properties of the material while cooling, the family of tensile curves shown in Fig. 7 was obtained, using 200-deg increments from room temperature to 1600 F. The decrease in modulus with temperature is of interest. The changing shape of the curves, showing quicker deviation from the modulus line, is also significant. Fig. 8 gives the summary of tensile strength, yield strength, and modulus over this temperature range. Yield strength is given at 0.2% and 0.02% offset, for information. Above 1400 F the iron has low strength, but as it cools there is a steady increase until it reaches approximately 600 F, at which

Fig. 5 – Row of holes in compressed-air pipe direct air at valve compartment to accelerate cooling of this part of casting

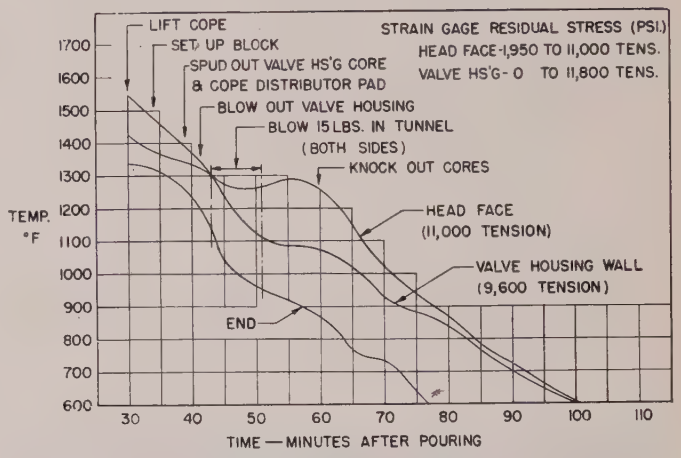

Fig. 6 – Cylinder-block cooling data – after cope lift – 1951 practice

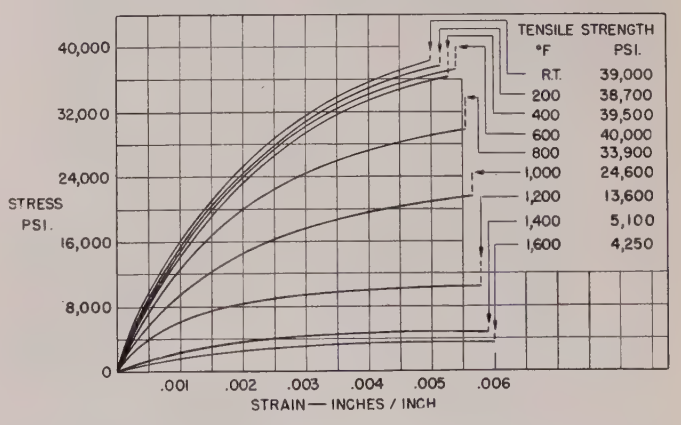

Fig. 7 – Cylinder-iron tensile tests – 0.005 in. per in. per min strain rate; 200-217 Bhn; 30 min at various temperatures ± 5 deg; 2-in. gage length, 0.505 in. diameter

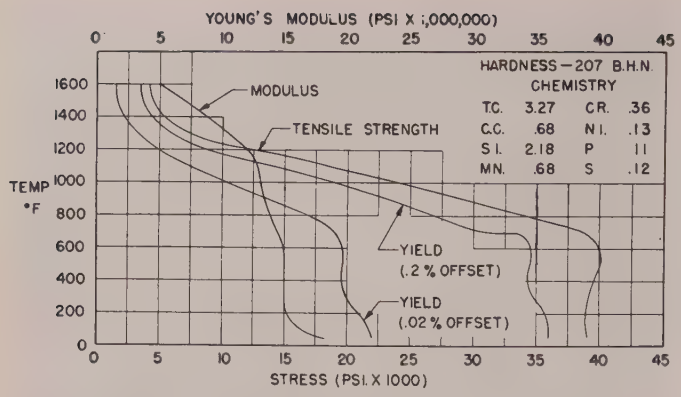
YOUNG'S MODULUS (PSI X 1,000,000)
0 5 10 15 20 25 30 35 40 45
HARDNESS — 207 B.H.N.
CHEMISTRY
T.C. 3.27 C.R. .36
C.C. .68 N.I. .13
S.I. 2.18 P .11
MN. .68 S .12
MODULUS
TENSILE STRENGTH
YIELD (.2 % OFFSET)
YIELD (.02 % OFFSET)
TEMP °F
1600 1400 1200 1000 800 600 400 200 0
STRESS (PSI X 1000)

Fig. 8 — Physical properties of cylinder iron

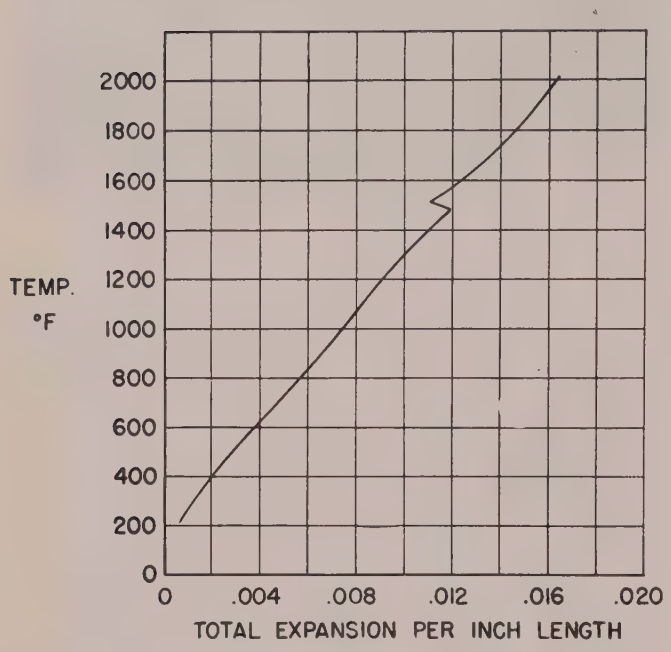

Fig. 9 — Cylinder-iron expansion — from room temperature to temperature indicated

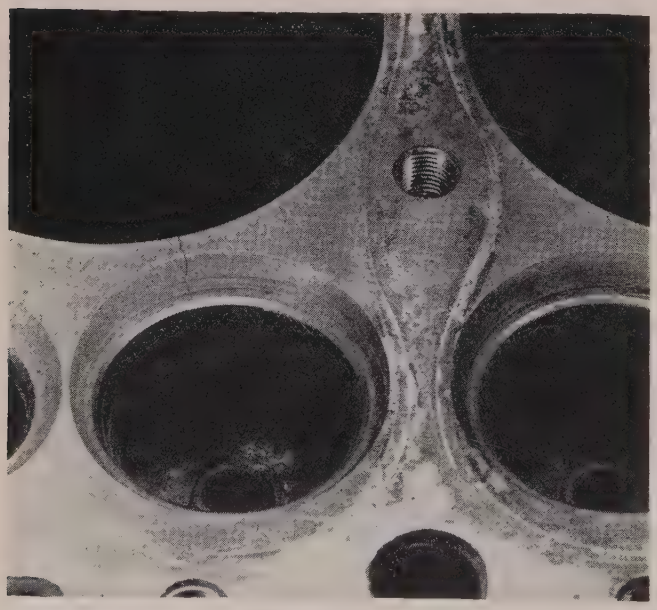

Fig. 10 — Typical crack, which occurred in nearly all cases between intake-valve seat and No. 4 or No. 5 cylinder

point it has attained substantially its ultimate strength.

From these data it is clear that cooling differentials between different portions of a casting may not be serious, in terms of eventual residual stress, at temperatures over 1400 F, because the hotter portion may yield and adjust itself to some degree. Severe temperature differentials over the steep part of the curve can cause cooling cracks, or leave residual stresses of varying degree, depending on the conditions.

For further information, the thermal expansion character of the same metal is shown in Fig. 9. Over the important range from 1400 F to 600 F, a casting 36 in. long must contract over ¼ in. This gives a further idea of the stress possibilities when different portions are cooled at different rates.

Application of the preceding information to the specific problem of cracks on cylinder-block top faces required still another tool. The percentage of cracks was very small, and we did not like to wait several months to see if a change in practice had eliminated the fatigue-type failures. The quantitative residual stress method gave us the tool to evaluate changes quickly.

The complexities of foundry operation also offer a challenge in such a program. In the Packard foundry there are two cylinder-block lines, which differ considerably in arrangement. The cooling cycle after the cope is lifted is quite dissimilar. There are also changes in conveyor speed, due to production levels, and there are interruptions at noon hour and at the end of the shifts. Sometimes breakdowns interrupt operations. Each of these items changes cooling conditions for a number of castings. Since it was not practical to rearrange the foundry and otherwise control these variables, it was necessary to develop methods for each line, at different conveyor speeds, that gave acceptable results.

Engineering Aspects of Problem

When the Engineering Research Division entered into the investigation of cracked cylinder blocks, a number of exploratory surveys were made. The distribution of the cooling water around the cylinders, particularly the center ones, was checked. Core changes were made to assure better circulation around the cylinders and intake ports. Stresses due to tightening the cylinder-head studs were determined at the point where the cracks occurred and were found to be compression stresses. Hot and cold cycling tests were run to attempt to crack the blocks without success, although approximately 9100-psi tensile stresses were found, due to this procedure. Chamfering and rolling the corner of the cylinder bore and top face were investigated. Hydraulic pressure in the combustion chamber was found to cause compressive stresses in the valve-seat area. Severe preignition tests were made on dynamometer stands without causing cracking. Hardness variation of the iron did not correlate with the strain variations found.

By this time enough service data had been accumulated to show that most of the cracks occurred at cylinders No. 4 and No. 5 and that most of the defective castings had been run off the casting conveyor line called our "A" line.

The previous work done to stop cracks in the valve chamber, our exploratory work, and the service data convinced us that the cracks were due to the combination of high tensile stresses induced by our foundry practice plus severe service conditions, even though we were not able to duplicate the exact combination. Fig. 10 shows a typical crack, which occurred in nearly all cases between the intake-valve seat and No. 4 or No. 5 cylinder. Since the trouble occurred in only a small percentage of the blocks, it was necessary to evaluate casting procedure very closely.

Up to now casting cracks had been eliminated by casting procedure changes, but no method had been developed to determine the amount of the trapped stresses left in the block even though it did not crack.

The value of these stresses was felt to be of extreme inportance so we could know how close we were to trouble even though no failures occurred.

The first procedure was of a general exploratory nature with stresscoat and drilled holes. The entire top face of the block was sprayed and small holes were carefully drilled through the brittle lacquer and the resulting cracks in the coating were noted. Radial cracks indicated tensile stresses, and circular cracks around the holes, that sometimes required a red dye to become visible, indicated compressive stresses, while no cracks meant no or very low stresses. Most of the new or used blocks showed tension at or near the points of failure.

The second method was to drill and accurately ream holes either side of the area in question, insert small drill rod pins, and carefully measure the distance between the pins by gage blocks. A saw cut was then made between the pins, a second measurement taken, and the change in distance determined.

This test gave quantitative data, but the results indicated more stress than could be possible, so the technique was abandoned. Later data indicated that the stresscoat method was more accurate than the pin method.

The method used next was application of SR-4 strain gages to the highly stressed areas, which gave fine measurements, good reproducibility, and reasonable strain data. Fig. 11 shows strain gages applied to a block. The procedure for mounting was as follows:

1. Thoroughly clean all surfaces where gages are to be mounted. Carefully hand-grind cast surfaces on area where gages are to be mounted. Mount the gage in the direction of the maximum strain.

2. Take gage reading before making the saw cut.

3. Make a saw cut 0.062 to 0.092 wide by 1 to 2 in. deep as close as possible to the free end of the strain gage. Fig. 12 shows a close-up view of the saw cut on the top of the block and Fig. 13 the saw cut in the valve chamber.

4. Take gage readings after making the saw cut.

5. Take the difference in gage readings, which measures the strain originally present in the 0.250 length of metal covered by the grid of the strain gage.

6. Where the metal was originally in tension, the saw cut permits the metal to contract to a value of zero strain. The gage actually measures the contraction, thereby showing a compressive strain equal to the original tension present.

7. The strain readings can be converted into stresses by multiplying by Young's modulus, which are the stresses present before the block is cut.

Actually, the total value of the stress or strain is not obtained by this method. We do not release all of the strains originally present, so we do not

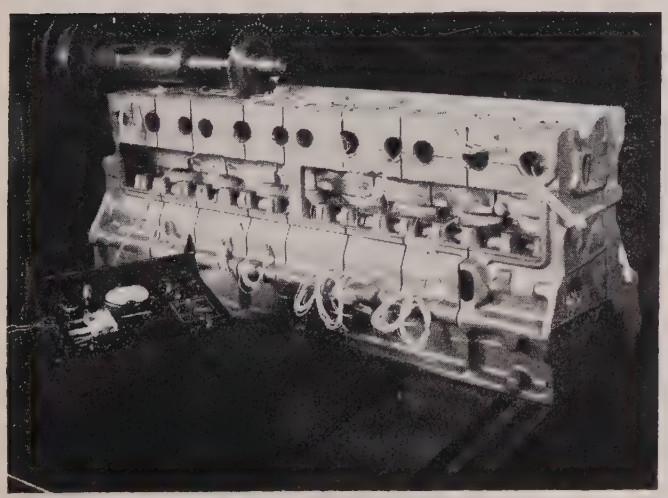

Fig. 11 – Strain gages applied to block

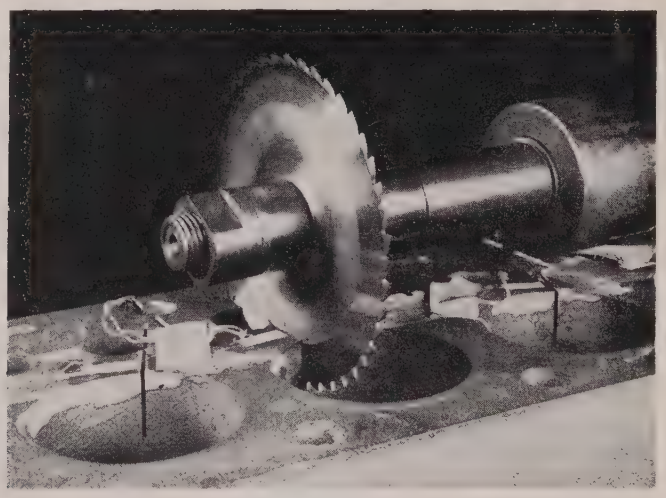

Fig. 12 – Close-up view of saw cut on top of block

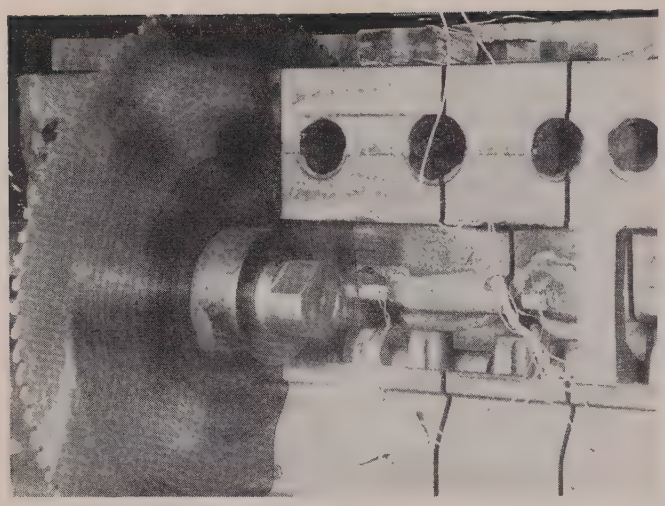

Fig. 13 – Saw cut in valve chamber

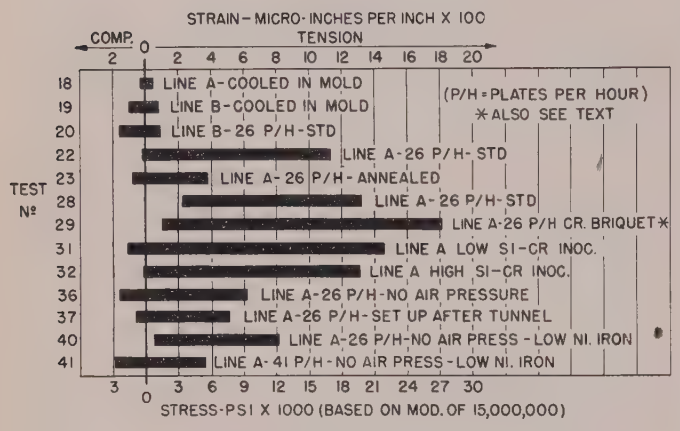
STRAIN – MICRO-INCHES PER INCH X 100
COMP.
TENSION
2 0 2 4 6 8 10 12 14 16 18 20
18 LINE A-COOLED IN MOLD
19 LINE B-COOLED IN MOLD
20 LINE B-26 P/H-STD
22 LINE A-26 P/H-STD
23 LINE A-26 P/H-ANNEALED
28 LINE A-26 P/H-STD
29 LINE A-26 P/H CR. BRIQUET*
31 LINE A LOW SI-CR INOC.
32 LINE A HIGH SI-CR INOC.
36 LINE A-26 P/H-NO AIR PRESSURE
37 LINE A-26 P/H-SET UP AFTER TUNNEL
40 LINE A-26 P/H-NO AIR PRESS - LOW NI. IRON
41 LINE A-41 P/H-NO AIR PRESS - LOW NI. IRON
(P/H = PLATES PER HOUR)
*ALSO SEE TEXT
TEST Nº
3 0 3 6 9 12 15 18 21 24 27 30
STRESS-PSI X 1000 (BASED ON MOD. OF 15,000,000)

Fig. 14 – Cylinder-block valve-seat strains – range of strains from minimum to maximum per block

actually arrive at zero strain, as described above. The part carrying the gage should be cut out of the block and all of the metal removed very carefully except a thin layer carrying the gage to get the strain value. We are working on techniques to establish the true stresses and we feel that we should find some values near the tensile strength of the iron. The values found, however, approach the true values, so they are very good indicators of the strain or stress present.

The data obtained indicated that the slower cooling of the top face permitted the balance of the block to contract ahead of the face, thereby causing tension in the top face as it attempted to shrink on cooling. Fig. 6, covering 1951 practice, showed slower cooling of the head than the valve housing. The problem was to cool the balance of the block slower but the top face faster, to eliminate the tension stresses at the point in cooling. It was necessary not to "overdo" the method or the trapped stresses would be "chased" back to the valve chamber or some other point of the block. The sum total of the tension and compression forces must be equal, the same as in carburized parts or other members carrying trapped stresses. Almen and others have shown diagrams indicating the necessary balance of forces.

Now that a method had been developed that gave a good indication of stresses, the cooling procedure was carefully changed on individual cylinder blocks and the results were measured.

Fig. 14 is a sheet from a report showing certain changes in foundry procedure and the stresses at the valve seats on the top face resulting from these changes. Test 29 is of particular interest. Investigation revealed this block stopped during noon hour in line with air jets directed at the valve chamber, resulting in excessive cooling at this point. The test really represents this cooling condition rather than addition of chrome briquettes. Fig. 15 shows variations in the air pressures in the pipes conducting air to the jets, blowing air on the block in the cooling tunnel, and the resulting variation in valve-seat stresses.

As a result of the series of stress tests on blocks (some 64 to date), the cooling tunnel air pressure on either side of the block was reduced to 0-5 psi, the valve chamber was thoroughly cleaned out by quick, high-pressure air blasts, the center of the top slab of sand continued to be broken up, and the peak line speed of 41 plates per hr was recommended.

It was definitely found that the stresses were produced by the handling procedures and timing on line A in removing the cores from the block and the block from the mold. Blocks cooled in the mold were free of stresses (but required ladle inoculations to maintain proper hardnesses and strength), and blocks made on line B were nearly stress free on the top face, due to the differences in line layout. Fig. 3 shows the uniform cooling rates of blocks cooled in the mold.

Fig. 16 shows original 1951 stresses in the block at the valve seats on the top face and final 1952 stresses at the valve seats and valve chambers after the corrective procedures were put into effect. Cooling curves have been taken on blocks made by present practice, but they do not differ from the previous curves sufficiently to account for the new results. More thermocouples would be needed to establish fully all the thermal gradients to account fully for the change.

These changes have been in effect for over a year. Service reports show the changes have eliminated practically all the cracks on the top surface to date, with very few cracks in the valve chamber or other suspected points due to trapped stresses.

We would not want to leave the impression that all casting troubles have been eliminated. We still have the general run of typical production troubles. One block was found cracked on the top face where the line had been shut down for noon hour and this block had air blowing into the valve chamber during the shut down as the air valve had been left on. The stress-relief practice of heating

blocks for annealing or preheating for welding may leave severe residual stresses, depending on cooling practice in the critical range. It now takes an unusual combination of circumstances to produce cracks.

We believe careful application of this method will take some of the art out of casting procedure and replace it by a means of determining with reasonable accuracy how far we are from trouble. Before this method was developed all that we knew was that we were or were not in trouble and if castings were not cracking we didn't know how close they were to cracking. Further investigation will help to determine how far our observed values are from true values so that for a standard cut a multiplier may possibly be used to establish the true stresses.

The "multiplier" idea should be considered only where past experience has given the investigator a good picture of the effect of other cuts in the vicinity of the stressed point in question. Cuts at the valve seats at Nos. 3 and 5 cylinders were found to have the effect of further reducing the residual stresses at the valve seat at No. 4 cylinder and likewise the cut at Nos. 4 and 6 cylinders will reduce the stress at No. 5. Originally, the pieces were cut out of the blocks until the effect of the other cuts was determined.

The "cut" method as described was used to save time and labor in the investigation as we were in production with this method, which is still laborious. The method was also better for handling of gages and pieces.

On a new investigation the safest procedure is to cut the parts in question out of the casting to be sure to get the major portion of the stresses present immediately.

On new casting designs we plan to investigate immediately all locations on the castings that may be suspected of having trapped stresses as soon as the first production castings are made.

Conclusions

The joint conclusions of the Metallurgical and Engineering Divisions are as follows:

1. The greatest single factor in inducing trapped stresses in castings is too great a difference in the cooling rates of different portions between 1400 F and 600 F.

2. To eliminate residual tensile stresses at a specific section it is necessary to keep the cooling rate of that section at least as fast as that of other significant sections that by their geometry and faster cooling could induce tensile stresses at the section in question. The last part to cool will be in tension.

3. It is possible to move trapped stresses from one section to another by overcooling. Corrective measures must be applied with discretion.

4. A method has been developed that gives quantitative results that serve as good index values of the trapped stresses present even in complex castings. This method can serve as a tool to determine reasonably accurately the benefits derived from a change in casting cooling procedure.

We hope this discussion and the method covered may be of value to foundries making castings and to engineers concerned with casting design and casting troubles, particularly those of an intricate nature.

Acknowledgments

Acknowledgment by the authors is made of the work done by the team that handled this project.

In the Metallurgical Division, Ed Coon of the metallurgical staff had general direction of the project. Dean Shiels, foundry metallurgist, did most of the actual foundry work, and was responsible for process control. Tom Moormann, of the high-temperature laboratory, conducted the physical property tests.

In the Research Engineering Division, Ed Nash followed the project, working with the foundry, metallurgical laboratory, proving ground, and stress laboratory. Richard Zenker, in charge of the stress laboratory, handled the strain-gage work and cooperated in developing the technique to determine the stresses. Warren Bopp had general supervision of the work done. Gale Sprague of the Automotive Division handled the cycling and preignition tests.

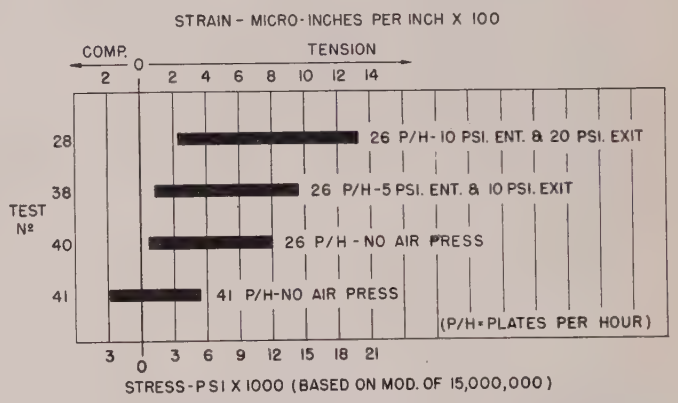
STRAIN - MICRO-INCHES PER INCH X 100
COMP.
TENSION
2 0 2 4 6 8 10 12 14
28
26 P/H-10 PSI. ENT. & 20 PSI. EXIT
38
26 P/H-5 PSI. ENT. & 10 PSI. EXIT
TEST Nº
40
26 P/H - NO AIR PRESS
41
41 P/H-NO AIR PRESS
(P/H=PLATES PER HOUR)
3 0 3 6 9 12 15 18 21
STRESS-PSI X 1000 (BASED ON MOD. OF 15,000,000)

Fig. 15 – Cylinder-block valve-seat strains – line "A" variations caused by air pressure changes

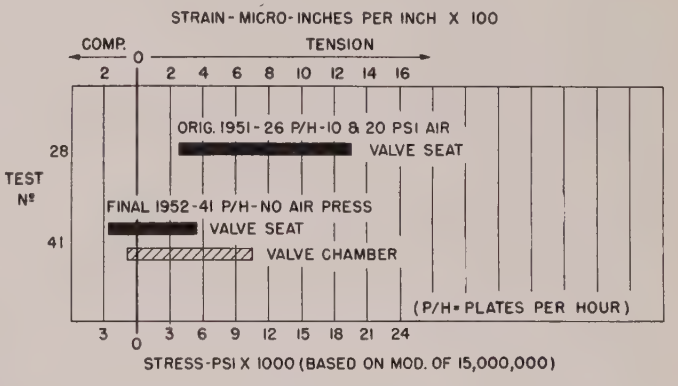
STRAIN - MICRO-INCHES PER INCH X 100
COMP.
TENSION
2 0 2 4 6 8 10 12 14 16
ORIG. 1951 - 26 P/H-10 & 20 PSI AIR
28
VALVE SEAT
TEST Nº
FINAL 1952 - 41 P/H - NO AIR PRESS
VALVE SEAT
41
VALVE CHAMBER
(P/H=PLATES PER HOUR)
3 0 3 6 9 12 15 18 21 24
STRESS-PSI X 1000 (BASED ON MOD. OF 15,000,000)

Fig. 16 – Cylinder-block residual strains – line "A" original 1951 and final 1952 blocks

Antiknock Requirements of

RESULTS of the two largest cooperative surveys made in 1951 and 1952 of the antiknock requirements of passenger cars are presented in this paper.

Conducted under the auspices of the Coordinating Research Council, data on more than 800 postwar cars are compared with previous surveys and are analyzed for relationship of car design, car age, engine speed, test conditions, test fuel type, make of transmission, and geographical car distribution to octane-number requirements.

■

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OF all the factors involved in the complex engine-fuel relationship, antiknock performance has been regarded as having a major influence on customer reaction. Because of the importance of this performance property, the extent to which their products operate knock-free in the hands of the public are of interest to the vehicle and fuel manufacturers. For this reason cooperative surveys of the antiknock requirements of cars in customer service have been among the best supported activities carried out under the sponsorship of the Motor Fuels Division of the Coordinating Research Council. The data obtained from these surveys have been used as guides or supplementary information by individual automotive and petroleum companies in planning the antiknock level of their products.

It was obvious from early work that the octane-number requirement of vehicles of the same make and model varied considerably. This was demonstrated clearly in the 1941 CFR Road Detonation Tests.[1] Because of these variations, it was necessary to use large enough car samples to permit statistical determination of the percentage of cars knock-free on a fuel of a given octane number. The number of vehicle tests required to obtain significant data in this manner exceeds the facilities of most individual laboratories. For this reason, cooperative surveys were conducted by both the American Petroleum Institute and the Coordinating Research Council on a relatively small scale prior to World War II.

The trend toward engines of higher compression ratio following World War II stimulated more comprehensive surveys in 1948 and 1949 than had been conducted previously. The 1949 tests, which were summarized by Best, Gibson, and Taylor before the SAE in 1950,[2] covered 260 postwar model cars distributed by make according to national registra-

Passenger Cars—1951-1952

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This paper was presented at the SAE National Fuels & Lubricants Meeting, Chicago, Nov. 6, 1953.

tion figures. These tests provided a sound basis for measuring the trends in octane-number requirements of postwar cars in subsequent surveys. They indicated further that requirements determined in terms of primary reference blends of isooctane and normal heptane tended to be from one to two octane numbers lower than those measured in terms of the Research octane number of commercial-type fuels.

In 1951, the Motor Fuels Division of the Coordinating Research Council undertook a broader passenger-car survey program having the following objectives:

1. To supply additional statistical information on the octane-number requirements of postwar cars.
2. To determine the octane-number requirements of models having new-design engines or transmissions.
3. To determine the effects of age and use on the octane-number requirements of selected car makes.

In this paper are presented the results of the work conducted in 1951 and 1952[3] comprising data on a total of 753 cars. The 1951 survey covered the three objectives with special emphasis on the statistical feature, while in 1952 the statistical survey was omitted to permit concentration on the new-design cars and the effects of age and use.

Test Techniques and Fuels

Test Techniques – The octane-number requirement data reported were obtained during acceleration under substantially level road conditions. Because knock in some cars is more severe at part throttle than at full throttle, it was necessary to investigate a range of throttle openings. The requirements discussed throughout this paper are the highest observed, whether they occurred at full or part throttle.

The full-throttle technique employed was that designated as "Research Technique for Determination of Octane-Number Requirements of Vehicles on the Road" (CRC Designation E-1-748). This technique is described in detail in both of the reports on which this paper is based.[3] It was designed to minimize disturbance of combustion-chamber deposits during the test work. Checks of maximum requirement at the beginning and end of the tests indicated that in only a small number of cars did any change in requirement due to the testing occur. In most of these cars the change was slight, and, therefore, it can be concluded that the technique was reasonably satisfactory in this respect.

Since no part-throttle technique had been standardized, each participant used the method of his choice.

Test Fuels – While primary reference-fuel blends of isooctane and n-heptane have been the most

[1] See SAE Transactions, Vol. 50, (October) 1942, pp. 458-464: "1941 CFR Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, W. M. Holaday, and C. B. Veal.
[2] See SAE *Quarterly* Transactions, Vol. 5, July, 1951, pp. 335-345: "Antiknock Requirements of Passenger Cars," by H. W. Best, H. J. Gibson, and J. E. Taylor.
[3] Complete data from both surveys are presented in the Coordinating Research Council reports "Octane-Number Requirement Survey, 1951" and "Octane-Number Requirement Survey, 1952."

Table 1 – Composition and Octane Numbers of Severity Reference-Fuel Blends

Blend No.	Composition			Average Laboratory Rating	
	SR-10	N-heptane	Iso-octane	Motor	Research
1-S-51	15.3	35.0	49.7	72.9	75.0
2-S-51	22.9	36.4	40.7	74.4	77.5
3-S-51	32.2	38.0	29.8	75.7	80.0
4-S-51	36.2	36.7	27.1	77.2	82.5
5-S-51	38.9	34.5	26.6	78.7	85.0
6-S-51	41.8	32.2	26.0	80.1	87.5
7-S-51	44.9	29.7	25.4	81.4	90.0
8-S-51	46.1	26.5	27.4	82.9	92.5
9-S-51	43.6	21.9	34.5	84.7	95.0
10-S-51	40.8	16.9	42.3	86.6	97.5
11-S-51	37.9	11.6	50.5	88.3	100.0

Table 2 – Identification and Octane Numbers of Commercial-Type Gasolines

Blend No.	Identification	Average Laboratory Rating	
		Motor	Research
1-C-51	RMFD 39-51	71.3	74.7
2-C-51	50/50 RMFD 39-51 and RMFD 40-51	73.7	77.6
3-C-51	RMFD 40-51	75.9	80.4
4-C-51	50/50 RMFD 40-51 and RMFD 41-51	77.6	83.1
5-C-51	RMFD 41-51	78.3	84.8
6-C-51	50/50 RMFD 41-51 and RMFD 42-51	79.4	87.3
7-C-51	RMFD 42-51	80.5	89.7
8-C-51	50/50 RMFD 42-51 and RMFD 43-51	82.6	92.5
9-C-51	RMFD 43-51	84.7	95.1
10-C-51	50/50 RMFD 43-51 and RMFD 44-51	86.2	97.5
11-C-51	RMFD 44-51	87.1	99.4

convenient standards for evaluating the knocking tendencies of engines and fuels, it has been found that the results obtained with these fuels do not always correctly reflect the antiknock performance of engines or gasolines as used by the public. For this reason, requirements were measured in terms of two series of fuels, severity reference fuels and commercial-type gasolines, in addition to the primary reference-fuel blends.

The severity reference fuels were prepared by blending diisobutylene (SR-10) and ASTM iso-octane and n-heptane in 2.5-octane-number increments to cover the range of Research Method octane numbers from 75 to 100. The sensitivities of these blends increased with octane-number level in a manner similar to the commercial-type gasolines described below. Blend numbers, composition, and octane numbers based on a large number of ratings are given in Table 1.

Six commercial-type gasolines covering the range of Research Method octane numbers from 75 to 100 in 5-octane-number increments were cross-blended to provide increments of approximately 2.5 octane numbers. The identification and octane-number ratings of these fuels, which were used in the 1951 survey only, are given in Table 2.

Inspection data for these fuels are presented in Table 3.

Discussion of Test Results

Statistical Survey, All Cars – In the 1951 statistical survey, data were submitted on a total of 450 postwar vehicles.

The following makes were represented:

Buick	Dodge	Frazer	Packard
Cadillac	Ford 6	Lincoln	Plymouth
Chevrolet	Ford 8	Mercury	Pontiac
Chrysler	Hudson	Nash	Studebaker
De Soto	Kaiser	Oldsmobile	Willys

The 1951 survey differed from the 1949 survey in that no less than 10 cars of each make were tested. This constituted an improvement, since it

Table 3 – Inspection Data, Commercial-Type Gasolines

Blend No.	Identification	Octane Number		API Gravity, deg	Rvp	ASTM Distillation Temp at % Evaporation				Tel, ml/gal	Sulfur, %	Bromine No.	Composition
		Motor	Research			10	50	90	Ep				
1-C-51	RMFD 39-51	71.3	74.7	60.7	8.4	133	240	330	384	0.30	0.016	20.8	61% th. reformate, 28% light st. run 6% aromatic solvent, 5% normal butane
3-C-51	RMFD 40-51	75.9	80.4	61.2	8.5	131	235	322	375	0.54	0.017	32.7	60% th. reformate, 21% light st. run 7% dib, 6% aromatic solvent, 6% normal butane
5-C-51	RMFD 41-51	78.3	84.8	59.6	7.8	141	236	358	414	1.59	0.037	....	66% th. reformate and th. cr., cat. cycle oil, 15% light st. run; 265 ep 13% fluid cat. cr.-heavy-420 ep 4% fluid cat. cr.-light-300 rp
7-C-51	RMFD 42-51	80.5	89.7	59.9	7.8	134	215	361	420	1.69	0.118	....	46% th. cracked and st. run, 6% th. reformed, 46% cat. cracked, 2% n-butane
9-C-51	RMFD 43-51	84.7	95.1	61.6	6.7	138	218	358	426	1.50	0.075	52.5	70% fluid cat. cr., 20% light alkylate, 10% light st. run
11-C-51	RMFD 44-51	87.1	99.4	60.4	6.7	144	227	371	428	3.05	0.086	70.7	75% fluid cat. cr., 15% light alkylate, 10% dib

Table 4 – "High-Requirement" Cars Tested

Make and Model	Years	Number Tested
Buick 50 and 70	1946-51	28
Cadillac	1946-51	28
Chrysler L-Head 8	1946-50	1
Oldsmobile V-8	1949-51	31
Packard	1946-51	9
Total		97

provided more representative data for cars comprising a small percentage of the total registration. The data were placed on a statistical basis by weighting the distribution of requirements for the individual car makes according to national passenger-car registration figures. The distribution curve of maximum octane number requirements in terms of primary reference fuels for the 450 cars in the 1951 survey is compared in Fig. 1 with the curve for the 260 cars in the 1949 survey. The fact that the distribution curves are practically identical indicates that, measured in terms of primary reference fuels, the octane-number requirements of the total postwar car population did not change between the 1949 and 1951 surveys.

As in the 1949 survey,[2] a comparison of the maximum octane-number requirements of cars located in the Eastern, Central, and West Coast areas of the country indicated no significant difference among the cars located in the three geographical areas.

The distribution of octane-number requirements over the speed range using primary reference fuels for the 450 cars of the 1951 survey is shown in Fig. 2. Fig. 3 indicates that maximum knock occurred most frequently with primary reference fuels in the speed range between 750 and 1400 rpm. It is of interest that 29% of the cars had maximum requirements at speeds above 1400 rpm, while 22% were in this range in the 1949 survey. This change may reflect the increased use of torque-converter transmissions which limit the minimum full-throttle engine speeds.

"High-Requirement" Cars – For comparison with the 1949 survey, data from cars for which the manufacturer recommends the use of premium fuels were analyzed as a group. The makes and models used and the number of each are shown in Table 4.

The distribution of maximum requirements in terms of primary reference fuels in Fig. 4 is within one octane unit of a similar distribution of requirements based on 64 "high-requirement" cars tested in the 1949 Survey.

New-Design Cars – Eleven car makes were selected for test as new-design cars to obtain information on new engine and transmission designs.

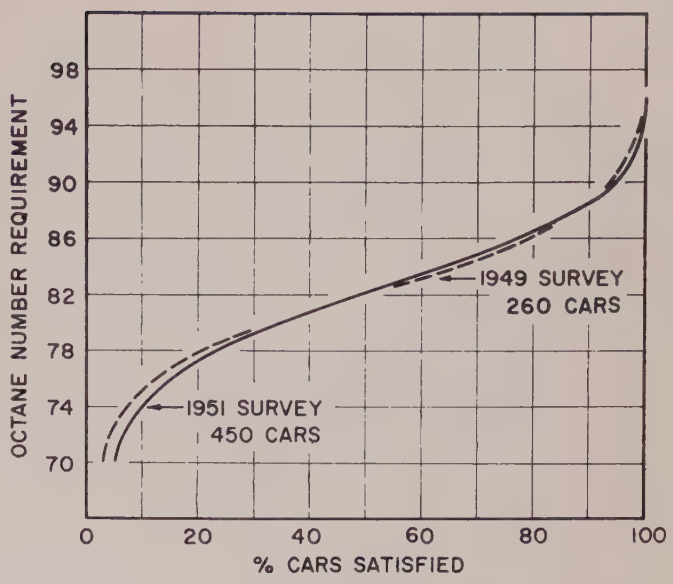

Fig. 1 – Distribution of maximum requirements – primary reference fuels

In the car selection, emphasis was placed on the new designs which might introduce significant changes in requirement.

The following makes and models were selected:

1952	Buick Series 70
1952	Cadillac
1952	De Soto V8
1952	Ford 6
1952	Lincoln
1952	Oldsmobile 98 or Super 88
1952	Pontiac 8, Hydra-matic
1951	Cadillac
1951	Studebaker V8
1950-1951	Chevrolet Power Glide
1951	Chrysler V8

A total of 98 of these cars was tested in 1951 and 303 in 1952. No less than 20 of any model were

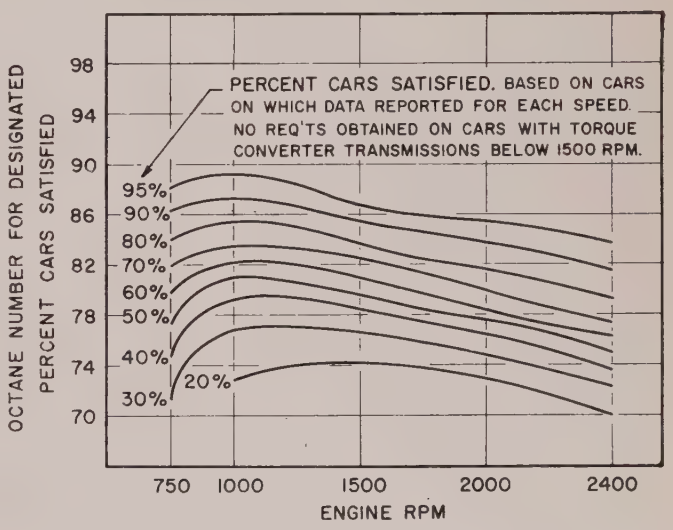

Fig. 2 – Distribution of requirements at various engine speeds – primary reference fuels – 1951 survey – 450 cars

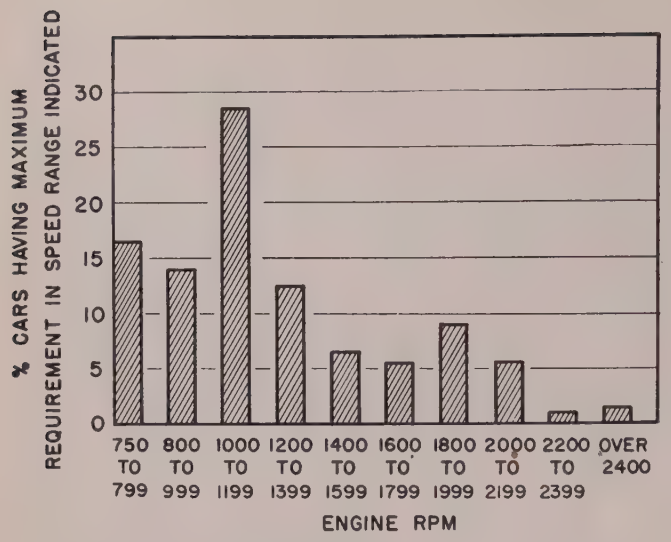

Fig. 3 – Speed for maximum requirement – primary reference fuels – 450 cars

Table 5 – Octane Number of Primary Reference Fuel for Indicated Per Cent Satisfied

Car	Survey	Number Tested	10%	50%	90%
B	1952	25	81	84	88
B	1951	28	79	82	85
D	1952	28	82	87	90
D	1951	23	79	83	87
G-1	1952	24	87	90	94
G-1	1951	21	85	89	92
G-2	1952	20	84	87	91
H	1952	29	85	89	93
I	1952	29	81	87	91
J	1952	28	79	84	90
L	1952	22	88	90	92
N	1952	30	79	83	86
N	1951	26	78	82	85
QA	1952	23	74	81	86
QM	1952	22	81	85	89
S	1952	23	78	83	89
Total		401			

used. The confidence limits for 95% certainty based on a 20-car sample are approximately ± 3 to 4 octane numbers between the 10% and 90% satisfied points.

The maximum requirements of individual car makes in terms of primary reference fuels at the 10%, 50%, and 90% satisfied points are given in Table 5. The spread in octane-number requirements between the 10% and 90% satisfied points for cars of the same make ranged from a minimum of 4 octane numbers for car L to a maximum of 12 octane numbers for car QA. Therefore, a given change in octane number will affect a greater percentage of cars of make L than of make QA.

At the 10% satisfied point the maximum difference in requirement among the car makes was 14 octane numbers. At the 50% and 90% points the difference was 9 octane numbers.

Effect of Age and Use – To obtain information on the effects of age and use on change in octane-number requirement, cars of the same model years were tested in succeeding surveys. With few exceptions, each car had at least 3000 deposit-miles. The average maximum octane-number requirements in terms of primary reference fuels and the average car mileages for each group of cars are shown in Fig. 5. The newer 1950 and 1951 models showed an increase in average requirement of from 1 to 3 octane numbers between the 1951 and 1952 sur-

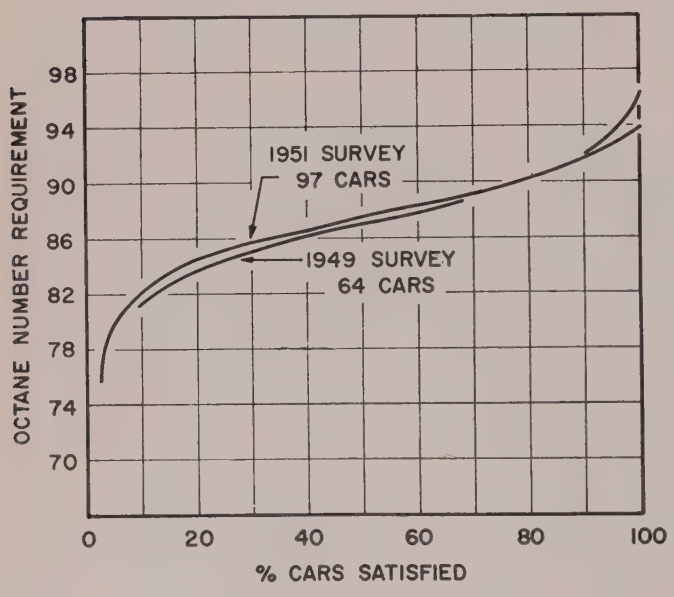

Fig. 4 – Distribution of maximum requirements for "high-requirement" cars – primary reference fuels

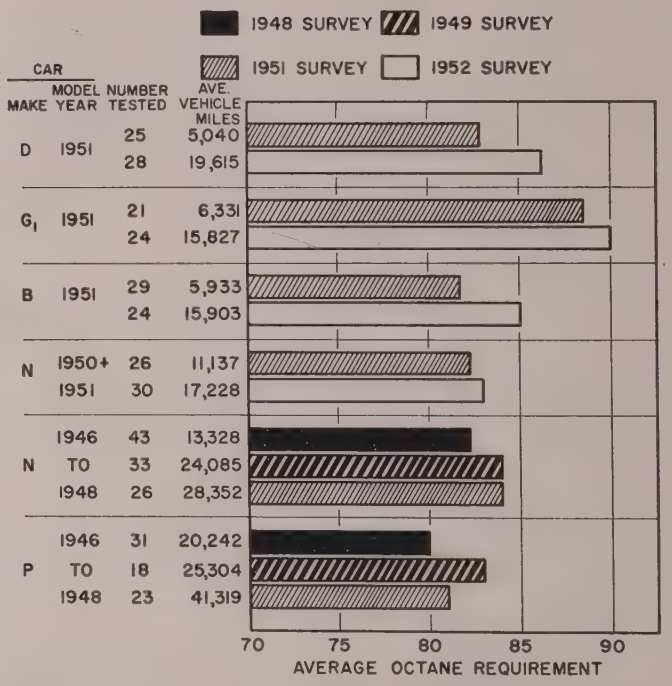

Fig. 5 – Effect of car age and use on average maximum requirements – primary reference fuels

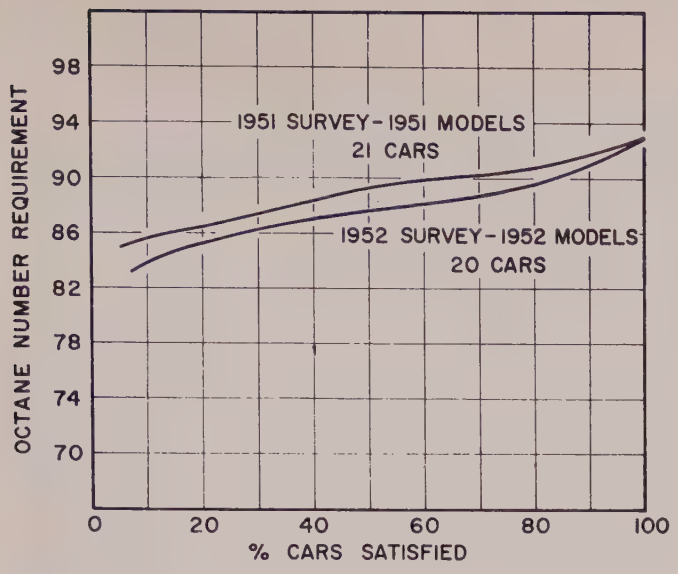

Fig. 6 – Comparison of maximum requirements of 1952 with earlier models of car G – primary reference fuels

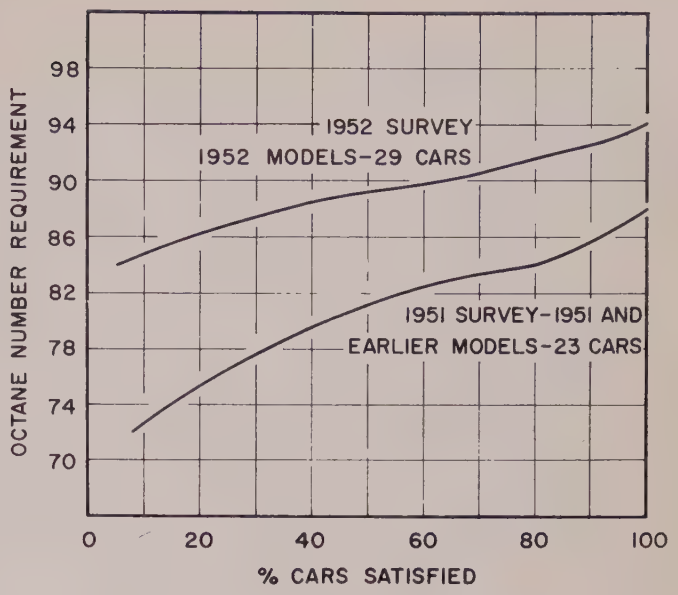

Fig. 7 – Comparison of maximum requirements of 1952 with earlier models of car H – primary reference fuels

veys. The 1946-1948 models of cars N and P increased in requirement between the 1948 and 1949 surveys. While these models of make N did not change in requirement between the 1949 and 1951 surveys, those of make P decreased 2 octane numbers.

These data indicate that after the initial 3000 miles of operation maximum octane-number requirements continue to increase for several thousand miles; however, no long-range predictions of requirement trends can be made.

Comparison of 1952 with Earlier Models – In Figs. 6 through 9, the maximum octane-number requirements of the 1952 models of four car makes are compared with the requirements of earlier models of the same makes as reported in the 1951 survey. With the exception of car G, the 1952 models have higher requirements than the earlier models of the same make. Due to differences in sample selection between the 1951 and 1952 models of makes H and L, the differences indicated in Figs. 7 and 9 may be exaggerated.

Effect of Automatic Transmission – The 1952 models of makes QM and QA were tested to determine the effect of one type of automatic transmission on requirements. The cars differed only in that car QM was equipped with a synchromesh transmission whereas car QA employed a torque converter. A comparison of the distribution of

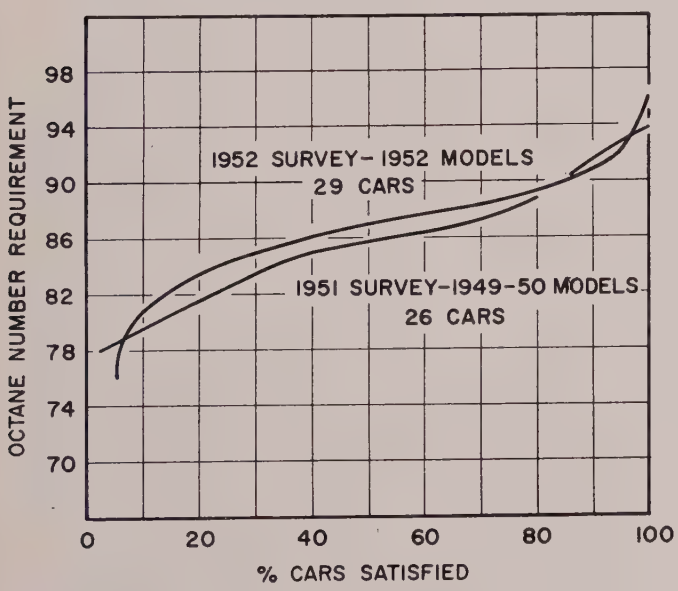

Fig. 8 – Comparison of maximum requirements of 1952 with earlier models of car I – primary reference fuels

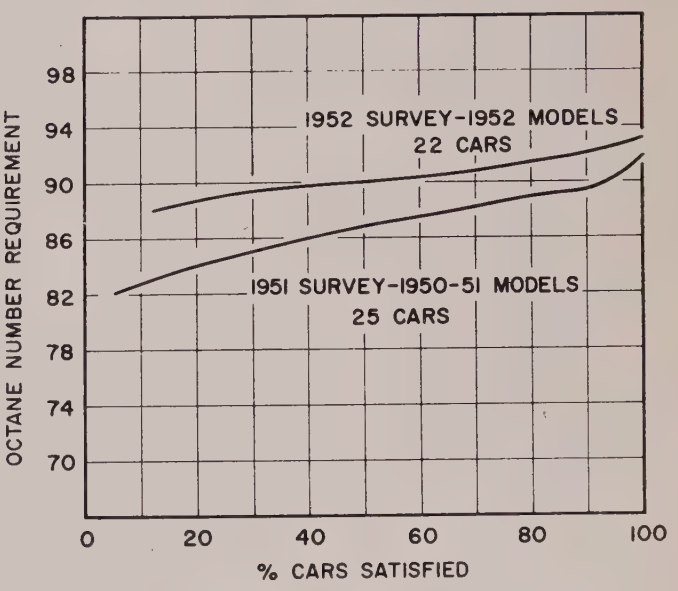

Fig. 9 – Comparison of maximum requirements of 1952 with earlier models of car L – primary reference fuels

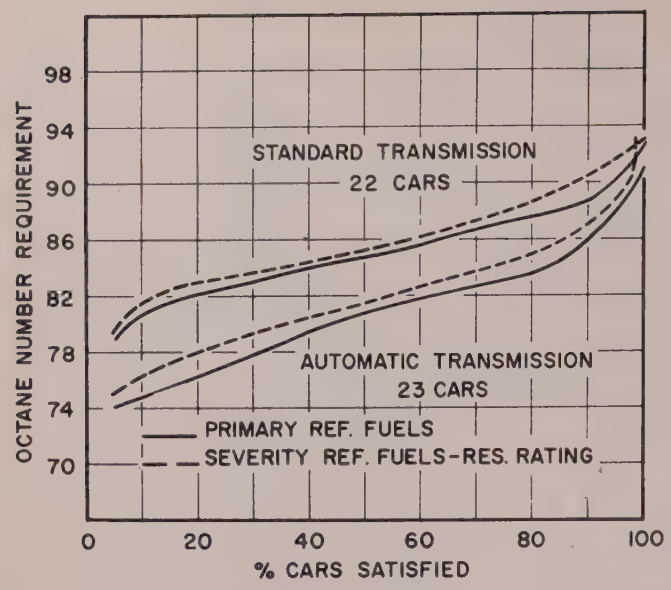

Fig. 10 – Effect of automatic transmission on maximum requirements of car Q

maximum octane-number requirements for primary and severity reference fuels is shown in Fig. 10. The maximum requirements obtained with cars using the torque converter were approximately 4 octane numbers lower than those obtained on cars equipped with the synchromesh transmission. This probably is due to the fact that the torque-converter transmission does not permit full-throttle operation at low engine speeds where maximum knock occurs with the synchromesh transmission. The effect of the torque converter in increasing the speed of maximum knock is shown in Fig. 11.

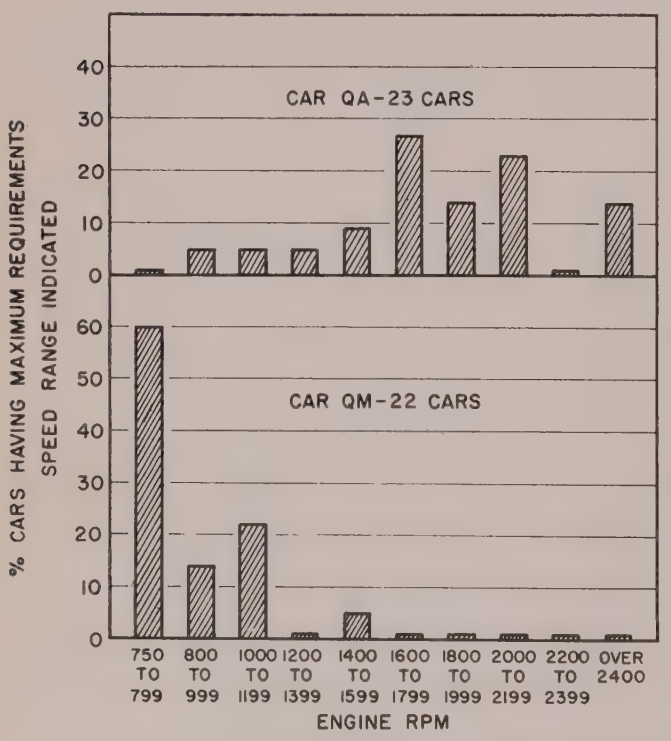

Fig. 11 – Effect of automatic transmission on speed of maximum knock – primary reference fuels

Table 6 – Per Cent Cars Having Maximum Requirements at Part Throttle

Car Make	Number Tested	%
B	25	0
D	28	11
G-1	24	17
G-2	20	70
H	29	31
I	29	59
J	28	46
L	22	27
N	30	0
QA	23	22
QM	22	59
S	23	57

Part-Throttle Requirement – There was considerable variation in the 1952 survey among the individual car makes in the percentages of cars having their maximum requirements at part throttle. The results for the individual makes recorded in Table 6 show that this percentage ranged from 0% for makes B and N to 70% for make G-2.

Tank Fuel Performance – Each car was first tested on the fuel in its tank to determine whether or not it knocked. In addition, the car owners were questioned as to whether knock was encountered on the tank fuel. The results separated according to grade of gasoline used are presented in Table 7.

The data for the three surveys are not directly comparable because of differences in car weighting between the 1949 and 1951 surveys and in car selection in the 1952 survey. It is interesting to note that the percentages of cars using regular-grade gasoline which were knocking has been consistently greater than the percentages found knocking on premium gasoline. It is also of interest that percentages of owners reporting knock are similar to the percentages determined by the observers.

Severity Reference and Commercial-Type Fuels – The distribution of maximum octane-number requirements obtained with the severity reference and commercial-type fuels are compared with those for primary reference fuels for the 450 cars in the 1951 survey in Fig. 12. In Fig. 13 the severity reference fuel distribution curve is compared with that using primary reference fuels for the 289 new design cars of the 1952 survey in which both series of reference fuels were tested. The commercial-type fuels were not used in the 1952 survey. Because of the differences in car selection, Figs. 12 and 13 are not directly comparable. However, as in the 1949 survey, the requirement curves for the severity reference and commercial-type fuels in both figures lie above the primary reference fuel curve, the maximum difference in the range from the 10% to 90% satisfied points being less than 2 octane numbers. The relative positions of the severity reference and commercial-type fuels are reversed in 1951 from their positions in 1949.

Figs. 12 and 13 are based on samples including

Table 7 – Per Cent Cars Knocking on Tank Fuel

Survey Year	Per Cent of Cars Found Knocking By Observers		Per Cent of All Cars Found Knocking	
	Premium Grade	Regular Grade	Observer	Owner
1949	35	54	48	38
1951	29	44	37	36
1952	33	42	35	31

many car makes. For individual car makes, the differences between the distribution curves for the primary and severity reference fuels may be considerably greater than two octane numbers as evidenced by the curves for make L in Fig. 14.

In summary, the results of our comprehensive program indicate that:

1. The octane-number requirements of the postwar passenger car population did not change between 1949 and 1951.

2. As in the 1949 survey, the 1951 and 1952 surveys showed no significant differences in requirement among cars located in the Eastern, Central, and West Coast areas of the United States.

3. Maximum octane-number requirements measured in terms of Research octane numbers of commercial-type and severity reference fuels for the car population as a whole are within 2 octane numbers of the requirements measured in terms of primary reference fuels. The differences in maximum requirements for individual car makes may be considerably greater, however.

4. Statistical survey data in 1949 and 1951 showed that maximum knock occurred most frequently in the speed range of 750 to 1400 rpm. The percentage of cars showing maximum requirements above 1400 rpm increased from 22% in the 1949 survey to 29% in the 1951 survey.

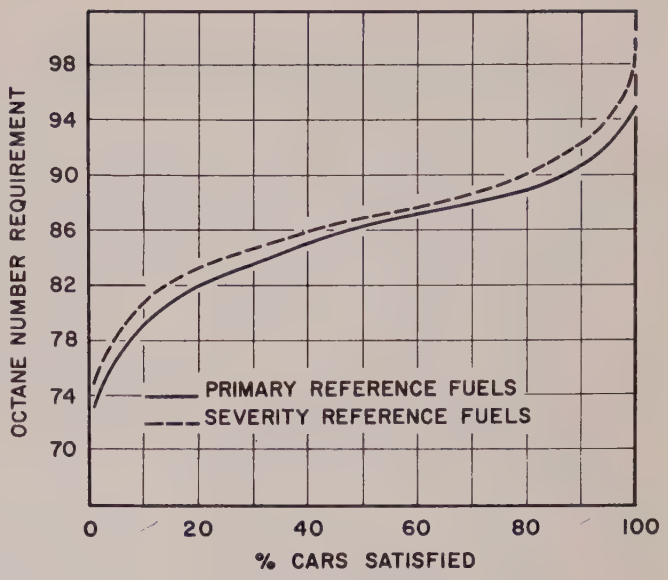

Fig. 13 – Distribution of maximum requirements of new-design cars – 1952 surveys –289 cars

5. The tank fuel data for the 1949, 1951, and 1952 surveys are not directly comparable, due to differences in car selection. It has been observed, however, that the percentages of cars using regular-grade fuel which are found knocking is consistently greater than the percentage for cars using premium-grade fuel. The percentages of cars found knocking by the observers have been similar to the percentages reported by their owners.

6. Limited data indicate that even after the initial 3000 miles of operation, maximum octane-number requirement continued to increase for several thousand miles.

7. A comparison of the individual makes tested

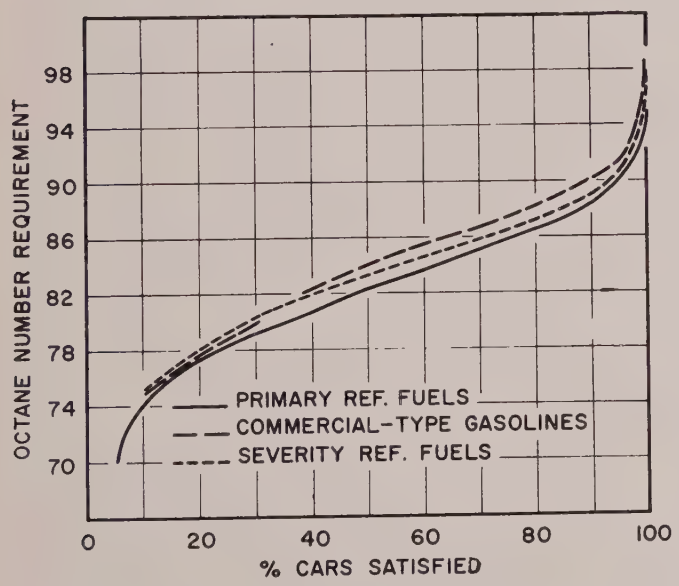

Fig. 12 – Distribution of maximum requirements –1951 survey –450 cars

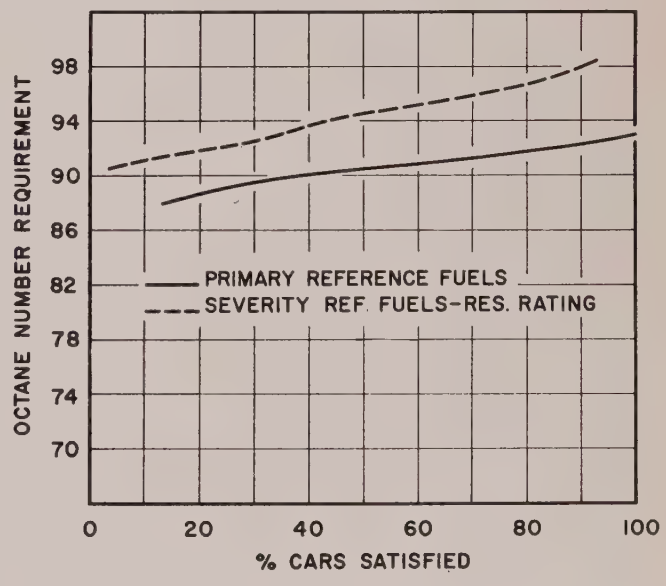

Fig. 14 – Distribution of maximum requirements of car L – 22 cars

in the 1952 survey showed a range of from 0 to 70% in the percentages having maximum requirements at part throttle. Maximum octane-number requirement was reported at part throttle in 11% of the cars tested in the 1951 statistical survey.

8. The maximum requirements for one 1952 make in which a manual shift transmission was used were approximately 4 octane numbers higher than for identical models using a torque-converter transmission.

The authors wish to emphasize that they are merely reporting the results obtained and analyzed by many technical men. All credit is due to them for their splendid cooperation and cannot be given to any individuals.

The organizations which contributed data were: Atlantic Refining Co.; California Research Corp.; Chrysler Corp.; Cities Service Research and Development Co.; Continental Oil Co.; E. I. du Pont de Nemours and Co.; Ethyl Corp.; Ford Motor Co.; General Motors Corp.; General Petroleum Corp.; Gulf Research and Development Co.; Pan American Refining Corp.; Phillips Petroleum Co.; Pure Oil Co.; Richfield Oil Corp.; Shell Oil Co.; Sinclair Research Laboratories, Inc.; Socony-Vacuum Laboratories; Standard Oil Co. (Ind.); Standard Oil Co. (Ohio); Standard Oil Development Co.; Sun Oil Co.; Texas Co.; Tide Water Associated Oil Co.; Union Oil Co. of California; and Universal Oil Products Co.

DISCUSSION

Discusses Accuracy of Octane Requirement Data

– J. E. Taylor
Gulf Research & Development Co.

MY discussion of this paper concerns the section on "New-Design Cars" in which the following statement was made: "The confidence limits for 95% certainty based on a 20-car sample are approximately ± 3 to 4 octane numbers between the 10% and 90% satisfied points."

Since I am not a statistician, I am not qualified to argue with this statement on the basis of a statistical analysis of the data; however, my interpretation of this statement is that on successive tests involving approximately 20 cars of a particular make and model, we might find a difference of 8 octane numbers in their octane requirement distribution curves. This is an alarming fact, and if true, we should run a much greater number of cars or quit octane requirement testing since the results are too questionable to be of much significance.

In an effort to develop some idea of the accuracy of octane requirement determinations, we made a very elementary analysis of the requirements of one make of car tested in the 1949 CRC Octane-Number Requirement Survey. The results of this analysis are shown in Fig. A. Make N was chosen because 50 cars of this make were tested. The maximum requirements of the 50 cars were tabulated without regard to any particular position and the distribution curve for the 50-car sample determined as shown by the heavy black line. Following this, the cars were split into four different groups of 25 cars each and the distribution curve determined for each group individually. These four groups were obtained by selecting (1) the first 25 cars, (2) the second 25 cars, (3) the odd-numbered cars, and (4) the even-numbered cars. The distribution curves for all five groups of cars are shown in Fig. A. Except at the very low level of "per cent of cars satisfied" the maximum spread is in the order of 2 octane numbers; and throughout the upper part of the curve, which is the part of primary interest, the spread is just over 1 octane number.

These data are reassuring in that they indicate that a 25-car sample can give a reasonably good picture of the octane requirements of a particular passenger car. We have analyzed a large amount of octane requirement data in a manner similar to that just described, and, although the precision is reduced as the sample size is reduced, we have never encountered a spread in requirements even approaching the 8 octane numbers indicated as a probability in the paper.

Esso's 1952 Survey Shows Higher Octane Requirement

– J. P. Hamer
Esso Laboratories Research Division

THIS paper constitutes an excellent summation and analysis of the very large amount of information obtained in the CRC Octane-Number Requirement Surveys for 1951 and 1952. Many petroleum companies attempt to obtain a picture of car requirements for their own purposes, but these generally represent only a small fraction of the scope of the CRC program discussed here. Along these lines, our laboratories have conducted periodic parking lot surveys to determine the octane requirements of the existing car population on the East Coast. This survey uses 100 cars statistically selected on the basis of the existing car population. It is encouraging to us that the results of our surveys, in general, are in very good agreement with the CRC survey data discussed in this paper.

Our surveys have shown one interesting deviation from the CRC survey results, and this is in the trend of octane requirement. The CRC Surveys for 1951 and 1949 are almost identical with respect to the octane-number requirement of the car population. Our 1950 survey is comparable in requirement level to both these CRC Surveys. However, our 1952 survey shows an octane requirement about 2 to 2.5 units higher at both the 50% and 90% car satisfaction levels than either of the other three surveys (1949, 1950, and 1951) mentioned. No explanation for this increase in our 1952 survey results is apparent to us.

We have also examined car octane requirements with increasing mileage, but like the authors, can reach no firm conclusion. Generally speaking, it appears that maximum requirements are reached somewhere between 15,000 to 20,000 miles. Thereafter, the mileage at which a decline in requirement will set in appears to be primarily a question of engine condition and more specifically of valve life.

Some Problems of Engine Design as Related to Fuel

– J. P. Charles
Pontiac Motor Division, GMC

THE authors of this paper have treated a complex relationship very objectively. Out of determinations of the engine-fuel relationship of 753 cars, they have pointed out

some very significant facts. I have particular reference to Figs. 6, 7, and 8 where the octane requirement effect of design changes in a given car make are shown. These comparisons are of particular interest to the car builder as they present data on the octane requirement of his cars in customer service that would be impossible for him to get through his own efforts. So, through CRC and the fine cooperation of its petroleum industry members, the car builder gets invaluable information to guide him in his engine development work.

Valuable as this information is, it does not tell the car builder how to design his new engines so that when they go on the market two years hence they will represent the ideal utilization of the commercial fuels of the day.

This he must do for himself, and rightfully so, just as the fuel refiner must accurately predict the future prevailing octane level when specifying refining processes to be used in his new refinery equipment.

Getting back to the car builder, let's take a look at some of his problems in arriving at a compatible relationship between engine and fuel.

The major specifications of his engine such as bore, stroke, combustion-chamber shape, and spark-plug location require about two years lead time to permit tooling. Specification of compression ratio and spark advance can be made sometime later but in reality have to be arrived at fairly close at the outset. This is because the car weight projected for the new engine calls for a certain value of engine torque to provide satisfactory performance. With the displacement and combustion-chamber shape already set, the compression ratio and spark advance will define the torque curve. So, if the car builder makes a serious error in setting his compression ratio too high and has to back off before going into production, the end result will be an underpowered car.

I will now list the principal items involved in bringing through a design where the car builder has only the prior knowledge of the art, his own experience, and common-sense judgment to guide him.

1. Prevailing Octane Levels Two Years Hence – If he uses intelligently all the information available, he can probably predict these within plus or minus one number, provided no national emergency arises.

2. Combustion-Chamber Design – Here the problem is to draw from past experience on other engines and from incomplete single-cylinder data to come up with the best compromise for smoothness, power, and octane requirement increase.

3. Combustion-Chamber Deposits – This is a big item, and the one on which a lot of effort is expended. The approach here lies entirely in the area of testing. The wrong approach will surely produce disastrously inaccurate data. The right approach is no guarantee of exactly the right answers. The reasons are well known. The relationship of deposits to octane requirement increase (ORI) is very complex. Starting with the clean engine the following considerations must be made:

(a) How many cars or engines in the program? It is no news that new engines that are ostensibly alike do not all have the same octane requirement. More than one must be used in the program, and the more the better.

(b) The selected compression ratio has manufacturing tolerances, and both extremes must be covered.

(c) Type of operating schedule for deposit accumulation has a great effect on ORI, probably as much as 6 to 10 numbers. Going too far in either extreme leads to strongly biased results.

(d) Lubricating-oil composition affects ORI by 3 or 4 numbers, so a judicious choice should be made here. Several types must be used in the program to cover the range.

(e) The Hill Factor is missed by those of us who do our development in the Detroit area, unless we put some of our cars in hilly areas for testing. That particularly bothersome kind of knock called wild ping has a way of rearing its ugly head on long grades.

(f) Altitude effect: The altitude at Pontiac is about 900 ft, so we must bear in mind that at sea level the octane requirement is one number higher.

(g) The Hooligan factor: This is the one we do not test for, the customer does it for us. When a product is produced in multiples in the hundreds of thousands, certain individuals driving certain examples of this product under certain operating conditions will come up with octane requirements that are a surprise to everyone. The percentage is small, but even one-tenth of one per cent of 400,000 is 400 cars, and we will hear from every one of them.

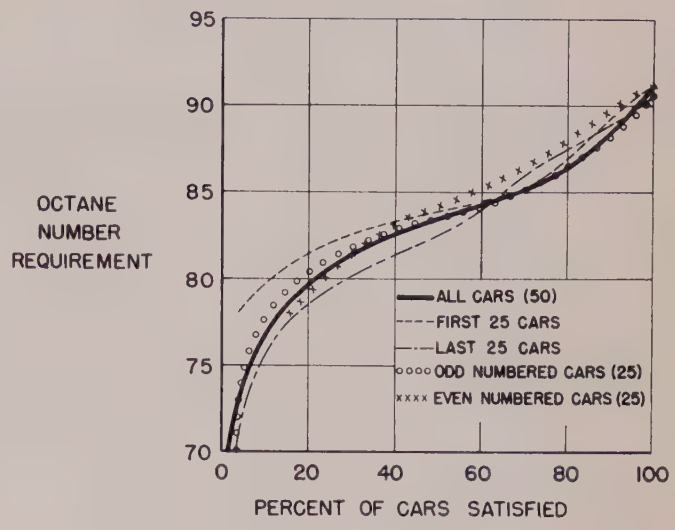

Fig. A – Effect of car selection on octane requirements of car N, 1949 CRC Octane-Number Requirement Survey

Authors' Closure To Discussion

WE will go back to the elements of statistics to answer Mr. Taylor. We believe the statement means that in 95 out of 100 samples of 20 cars each, the maximum spread between the curves will be ±3-4 octane numbers. In the few cases tested by Mr. Taylor, the spread was smaller than the values given. This is to be expected since, statistically, a large percentage of the 20-car samples will differ from the true curve of the car population by a smaller value than ±3-4 octanes. It would be necessary to test a larger number of samples than he did before the deviations encountered in practice can be safely compared with the values derived by the application of statistical theory. The specific magnitude of this confidence limit is of considerable importance in a program where the number of cars is limited, since it determines the number of cars that should be tested.

We can think of no explanation for the difference between the results reported by Mr. Hamer and the CRC results. His data showed an increase in the 90% car satisfied level of 2-2.5 octanes in his company's 1952 survey over their 1950 survey, whereas the CRC surveys showed no change in level between 1951 and 1949. This might be another indication that a sufficiently large sample of cars properly allocated is essential to a satisfactory statistical survey of the antiknock requirements of cars in customer service.

A Production Application of

AIRCRAFT structural engineers have long been intrigued by the possibility of eliminating the thousands of assembly operations involved in a typical airframe structure – one composed of many formed pieces joined by rivets and bolts – and of replacing these multiple operations with a double, or even a single, automatic operation employing simply temperature and pressure. However, it required the challenge presented by the guided missile, which calls for a more efficient structure in the case of thin sections, very high rates of production, lower costs, and small, easily expandable facilities, to advance this idea from a laboratory procedure to a production reality.

Our company has reached this goal on Air Force and Navy projects, through the use of adhesives to join metal skins to low-density cores, thus forming sandwich-type structures, and to replace rivets and bolts in attaching one metal part to another. The use of both of these adhesive applications in the construction of a single part has led to major savings of weight and to lower manufacturing time and cost.

The low-density core is honeycomb – a material that is itself fabricated by joining corrugated aluminum foil with adhesive to form hexagonal cells. The same adhesive is used throughout. Applications have been developed touching several projects, the most advanced of which is the B-61A Matador. (See Fig. 1.)

The Adhesive

The bonding agent we use is sold under the trade name of FM-47 by the Bloomingdale Rubber Co. It is a development of an earlier vinyl-phenolic thermosetting adhesive known under the trade names of FM-45 and Plycozite. Development of this adhesive was started in 1942 at Martin, in connection

USE of adhesives to join metal skins to low-density cores results in major weight, manufacturing time, and cost savings, according to the authors.

They report here that such sandwich structures have been used in several projects, the most advanced being the B-61A Matador.

To achieve a bonded joint between two aluminum sheets or between sheet and core, four generally similar steps are required. The sheet must be cleaned, coated with adhesive, force-dried to remove volatile materials, and finally cured under temperature and pressure.

The authors conclude that production adhesive bonding of flat and contoured primary structures, with all its inherent advantages, is now available to the industy.

The Authors

GEORGE E. HOLBACK is supervisor of the Materials Laboratory, Glenn L. Martin Co. He received his bachelor's degree in aeronautical engineering in 1938 and his master's degree in 1939 from New York University. Mr. Holback is currently engaged in developing improved materials for aircraft and guided missiles, and in work related to structures and their testing.

JAMES L. BURRIDGE is supervisor of structures projects, Glenn L. Martin Co. He received his bachelor's degree in aeronautical engineering from the University of Cincinnati in 1943. Mr. Burridge entered the Structures Department of the Martin Co. after graduation and since 1945 has been concerned with structures of special weapons and missiles.

Structural Adhesive Bonding

George E. Holback and James L. Burridge,
Glenn L. Martin Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

with an earlier investigation of sandwich structures. FM-47 is available today, both in liquid form and as a prepared film – from which part of the solvents have already been removed. The film derives strength for easy handling from its carrier of open-weave Fiberglas cloth. The threads in the cloth are spaced at approximately 0.1 in.

To achieve a bonded joint between one aluminum sheet and another, or between sheet and Honeycomb core, four generally similar steps are required. The sheet must be cleaned, coated with adhesive, force-dried to remove volatiles, and finally cured under temperature and pressure.

Cleaning

Cleaning is accomplished by first dipping the skin into an alkaline cleaner. Commercial examples are Sprex-AC, Oakite-6, and Kelite-14. This is followed by a water rinse and check for water break. Next, the sheet is given either a chromic-sulfuric bath for 10 min, followed by water rinse and drying in a 200 F oven, or a 15-min anodizing treatment without the sealing operation. Two secondary cleaning methods are used, when it is not expedient to dip the part. The part may be degreased by a 15-min bath of trichloroethylene vapor, followed by sand blasting with No. 30 grit. The other alternate method consists of scrubbing with phosphoric acid (trade name Deoxidine No. 624), then rinsing with cold water and checking for water break. No cleaning of the core is required.

Coating

Coating the parts with adhesive is essentially a problem of depositing sufficient glue-line thickness. This requires application of liquid adhesive as a first or primer coat. Additional thickness may then be built in terms of several subsequent coats of liquid or by application of the film. The liquid may be applied to sheet by spray gun, using FM thinner – in the ratio of 1½ to 1 – to cut the adhesive. For application to the core, a hand brush or roller-coater gives best results. Flat cores may be coated by an automatic roller.

Film is equivalent to approximately four coats

Fig. 1 – B-61A Matador

Fig. 2 – Use of button technique to measure qualitatively tensile strength of core-to-skin bond

of the liquid adhesive and may be applied to either surface, as may prove convenient. It may be attached to the part by laying the film on the primer coat, while it is still tacky, or by heating the part already primed to about 200 F – so as to produce tackiness. If the entire glue-line thickness is built up by liquid coats, half an hour of drying time is required between coating.

Drying

Drying of the adhesive before joining the parts is required to remove solvents. This may be accomplished by long periods at room temperature or accelerated by a higher temperature in an oven. Two hours at room temperature is adequate for prime coats of adhesive; however, 16 hr at room temperature is required for glue lines built up from several coats of liquid An additional force dry of 1 hr at 225 F is used for metal-to-metal joints for material applied as either film or liquid. This force dry is not required for core-to-face joints if film is used, except where strength is critical at elevated temperatures.

Curing Cycle

A curing cycle of 30 min at 300 F with sufficient pressure to hold the faces of the joint in good contact is required to complete the bond. An ideal cycle would consist of contact pressure during preheating to 300 F, breathing, and then sufficient pressure during the 30 min at 300 F to assure good contact. The amount of pressure required is a function of the thickness of the metal parts, the fit of the parts, and of the pressure required to produce flow in the adhesive. A practical compromise – for bonding methods other than that of using a platen press – is to apply a pressure of 15 psi to skin and core areas and of 100 psi to metal-to-metal areas throughout the curing cycle. The cycle is characterized by a momentary reduction of pressure over the metal-to-metal points at the time 300 F is reached. This momentary reduction, which allows escape of the remaining volatiles and produces a solid glue line, is called breathing.

Adhesive Strength

Using the process outlined, an adhesive joint between core and face material will have adequate strength to stabilize aluminum faces against buckling up to the compression yield. Tensile strength of the core-to-skin bond can be measured qualitatively by the button technique illustrated in Fig. 2. Metal-to-metal joints will typically develop over 4000 psi in the adhesive, when a standard ½-in. lap-joint specimen of 0.064 sheet is tested. The load that a given shear joint will carry is a function, primarily, of the stress in the metal sheet as it enters the bonded joint, of the length of the lap, the eccentricity of the joint, the yield strength and modulus of elasticity of the adherents, and the modulus of rigidity of the adhesive. Joint strength of FM-47 has not been found to be a function of glue-line thickness at room temperature, over a practical manufacturing range of 0.003 to 0.030 in. For a joint between two members of equal thickness and similar material an average entry strain or stress can be plotted against lap length divided by member thickness squared. This relation can be developed from theoretical considerations, if the physical properties of the adhesive and of the members are known.

Similar theory can be extended to give joint strength for members of unequal thickness and dissimilar materials. Fig. 3 is a nondimensional plot of joint entry stress, against the square of the ratio of lap length to sheet thickness, as determined from laboratory tests of samples of equal sheet thicknesses. Curves for 24ST, using both FM-47 and FM-45, are presented for comparison of the adhesives, and an additional curve for 75ST with FM-47, for a comparison of alloys. It is apparent that increasing the lap length reaches a point of diminishing return, beyond which no advantage is to be gained. These data provide the basic information for the detail design of a given joint, when proper factors of safety are introduced to allow for differences existing between production and laboratory techniques.

The temperature range of the adhesive, which complies with MIL-A-8331, gives values for ½-in. lap specimens as shown in Fig. 4. It is to be noted that there is a certain area beneath the psi curve. With the basic chemical composition of the adhesive unchanged, this area may be shifted up or down the temperature scale by adjusting preheat, curing temperature, and time.

Characteristics of the adhesive with regard to creep and fatigue have been investigated sufficiently to justify the statement that few limitations on its

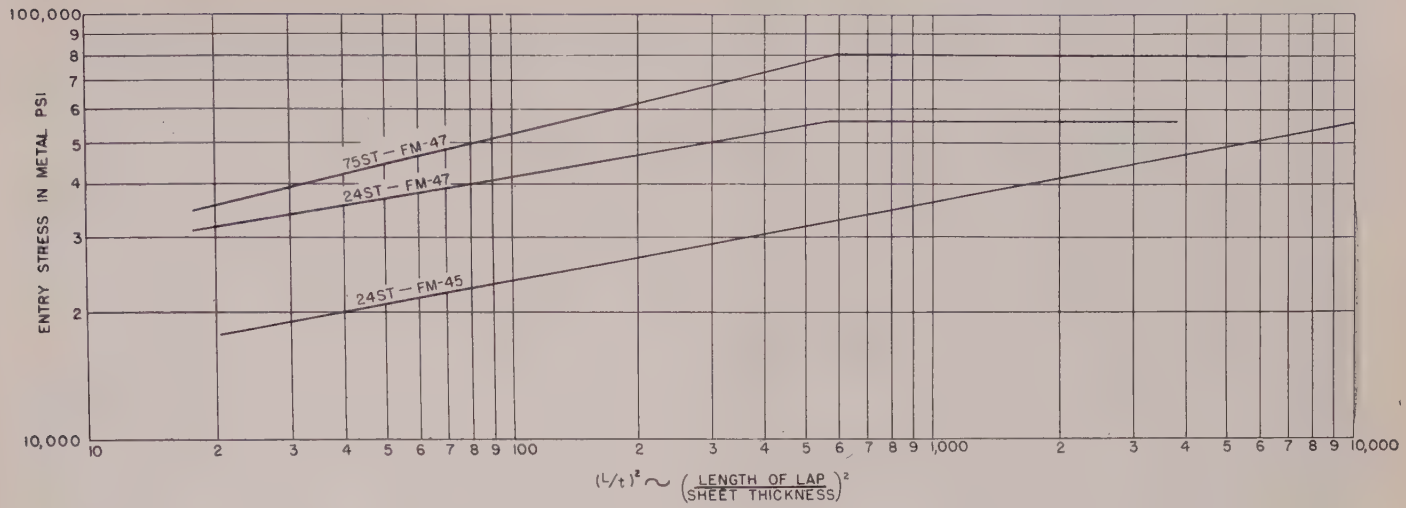

Fig. 3 – Nondimensional plot of joint entry stress versus (lap length/sheet thickness)^2

use from these standpoints are apparent, within the recommended temperature range of the adhesive. Humidity effects are not significant in reducing the strength of the adhesive. An incident that occurred on an early missile project demonstrates this. The missile, which had a sandwich-type wing, was flown for shipboard target practice off Chincoteague, Va. About a year later it reappeared in a fisherman's net off the coast of Atlantic City. In due time, the wing was returned to the company and samples were taken from it for column compression, flexure, and tension testing. The strength of the samples was good.

Application

The experience and background knowledge that made the use of adhesive on a broad scale possible in the Matador were gained on several earlier projects. The Gorgon IV, a pilotless aircraft, employed the first flight surface of sandwich construction – and approximately 80 wings were manufactured. No metal-to-metal bonding was involved. The Martin 2-0-2 and 4-0-4 used structural honeycomb floors consisting of flat sandwich panels.

Considerable development work was done on sandwich panels for use as cargo floors. Next, came the ISSM – the experimental version of the Matador – which used sandwich-type surfaces throughout, as well as flat panels for tank floors and for some bulkheads. A production version of the Gorgon IV, known as the KDM-1, followed – embodying the first major use of metal-to-metal bonding. The right and left wing panels were joined at the centerline by gluing the wing cover-skins to an aluminum casting. These applications, all involving production in some volume, paved the way for the

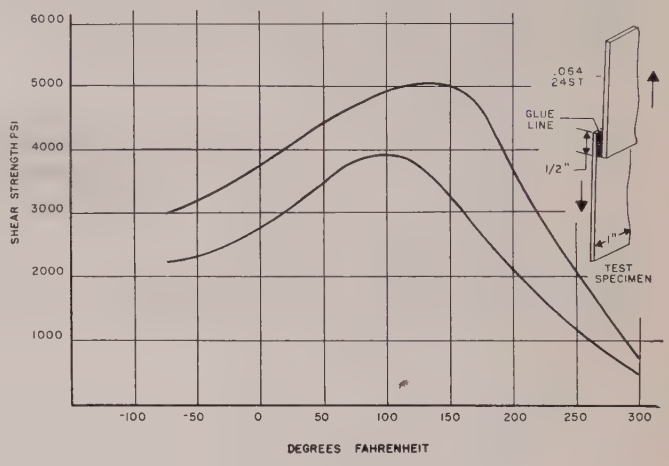

Fig. 4 – Variation of shear strength with temperature for FM-47 – minimum cure at 300 F

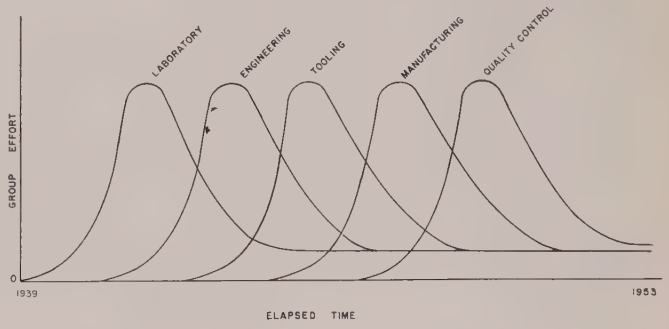

Fig. 5 – Efforts of various groups

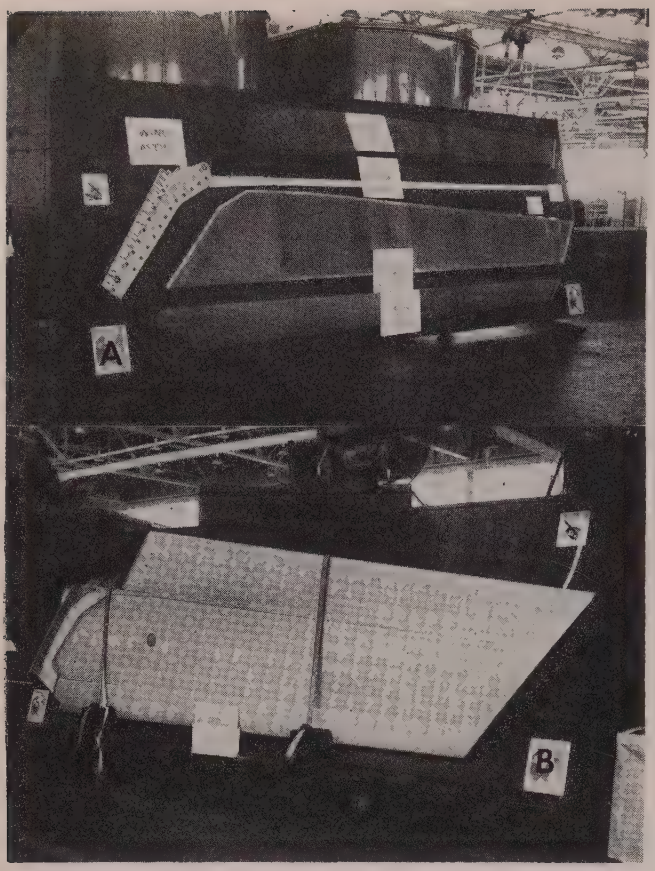

Fig. 6 – A: Wing components minus skins, before bonding. B: Complete wing

B-61A. Following the B-61A has come the developmental use of honeycomb bonded blankets as compression covers for the tail of the P5M-1 Martin flying boat.

In all these applications, it has been found that the production reality of adhesive bonded structures requires the application of sustained effort and the accumulation of know-how by five distinct groups, in successive waves of activity involving the materials laboratory, engineering design, tool engineering, manufacturing, and quality control, as seen in Fig. 5. Let us consider these waves of effort in order:

A. The laboratory effort has been under way for sometime, our company having begun its investigation of low-density core material before Pearl Harbor. Much has been written on the subject. Many of its phases are now being investigated under various Government contracts. Hence, it will not be discussed at length here.

B. As the laboratory effort passed its peak, the engineering design effort increased. The lessons learned by this wave of effort were, primarily, as follows:

1. A successful adhesive bonded design must be designed for adhesive at the outset. It must not be a conversion of a riveted design. Several reasons for this are apparent.

(a) Adhesive joints do not exhibit the nonlinear deformation characteristics of riveted and bolted joints, which were used in the past to reduce peak loads from shear lag and other effects. Thus, special consideration must be given to areas where loads tend to peak on the basis of linear moduli.

(b) Material laps for bonded joints are not necessarily consistent with edge distances for bolts and rivets.

(c) Higher sheet efficiencies may be attained by bonding.

2. The designer must become familiar with and consider in greater detail the tooling problems with the tool engineer so that the configuration may be reproduced reliably and cheaply. For example, expansion of the materials during the heating cycle must be considered. This becomes a major problem in bonding materials having different expansion coefficients. Also, the ability of the structure to withstand the pressures applied at the bonding temperature by the bonding tool must be considered.

3. Full advantage may be taken of the fact that once an assembly is prepared for a bonding cycle – the application of heat and pressure – it costs practically nothing to complete the major splices at the same time. This holds true because it is even cheaper to clean and spray adhesive on the whole part than to mask it for a subsequent operation. In most cases, the tooling may be affected only locally, since it is usually necessary to heat the whole part anyway. On the B-61A wing the front and rear spar, as well as wing-to-casting, splices are effected while the wing covers are being bonded to the core. Fig. 6A shows components minus skins, before bonding. Fig. 6B shows a complete wing.

4. Investigate all cases of highly loaded thin sections for adhesive bonding. Wings or aerodynamic surfaces made with a solid core – honeycomb filling the entire internal volume of the airfoil – are economical, in terms of weight, below a certain size, about 200 sq ft in area for surfaces 10% or less thick. Surfaces within this range have less weight because thin cover gages may be stabilized to a high working stress with a light core material. The size limitation is partially due to the scale effect of core weight, which is proportional to volume, increasing faster than the area. As size increases, more core weight is required. It also becomes possible to stabilize thicker cover skins to high stress levels in other ways, such as by using adhesive-bonded double-skin panels as covers. (See Fig. 7.)

5. Use standard sheet stock to make laminated tapered or sculptured skins. Wing cover skins tapered with standard sheet stock offer almost unlimited variations in thickness, at the same time greatly reducing the procurement problem. Doub-

lers may easily be added for extra bearing material or for local reinforcement.

6. Design for a breathing cycle in the case of all metal-to-metal applications, to make sure of solid glue lines. The stability of different parts, under various bonding loads must also be considered.

C. The tooling effort was started during the early part of the engineering phase, because of the acute interdependence of these activities. Tooling know-how had to be developed. Here are some of the important lessons that were learned:

1. Tools must be designed that assure the application of the required temperature and pressure *at the glue lines.*

2. To effect a short heatup time and good temperature control, the part – and not the entire tool – should be heated. Account must be taken, however, of the relative expansion between part and tool. Except for simple flat work which is usually done in a steam-heated press, this cycle is best accomplished by electric-blanket and pressure-bag bonding, which permits breathing, lower temperatures in the pressure bag, and consequently longer tool life.

3. For high rates of production and small, easily expanded facilities, electric-bag-diaphragm bonding should be investigated. A complete stabilizer, including left- and right-hand sides and allowing for the time required to join these to the as-cast surface of the centerline casting, can be assembled and made ready for inspection in 1 hr and 25 min. This can be achieved with extremely simple details, all that is needed being skin, doubler, casting, and core. All that would be required to operate these tools at another facility is a source of electric power and air pressure. (See Fig. 8.)

4. Cutting the contour to tolerance, thereby eliminating ribs and many contour tools for the spar and other details, poses no problem. (See Fig. 9.)

5. Mild forming or rolling of laminated sheets can be accomplished after the bonding process is complete.

D. The development of the manufacturing know-how proved a little more difficult. There are a great many people involved in the processing, and the transition from the drill-and-rivet system to a glue system is necessarily slow in the case of aircraft workers. Damage done by factory people is not always obvious at the time. Perhaps the most important watchword for those engaged in this activity is "cleanliness," and every individual contributes toward the objective – from management down to helper. For example:

1. It is imperative that all facilities be located in one area, with one man responsible.

2. Cleanliness cannot be maintained when too much handling takes place. This also increases

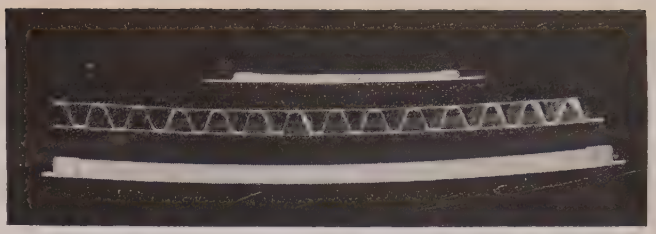

Fig. 7 – Adhesive-bonded double-skin panels for use as covers

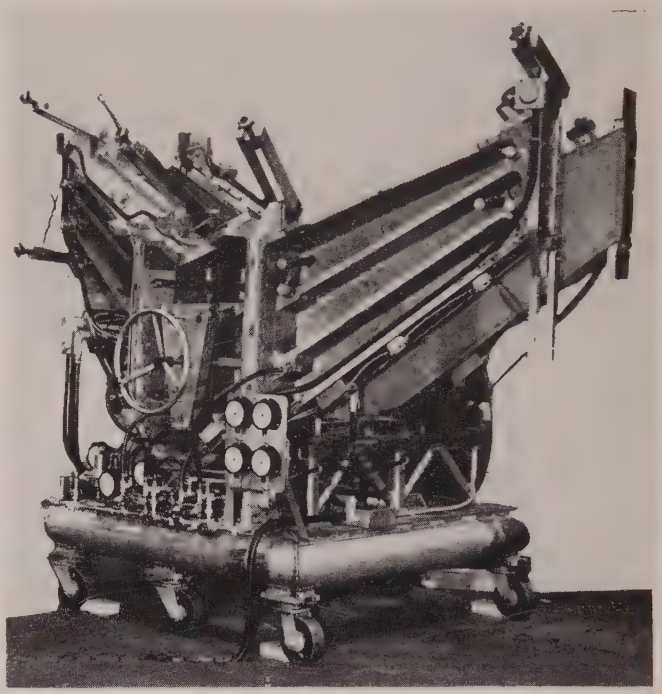

Fig. 8 – Source of electric power and air pressure

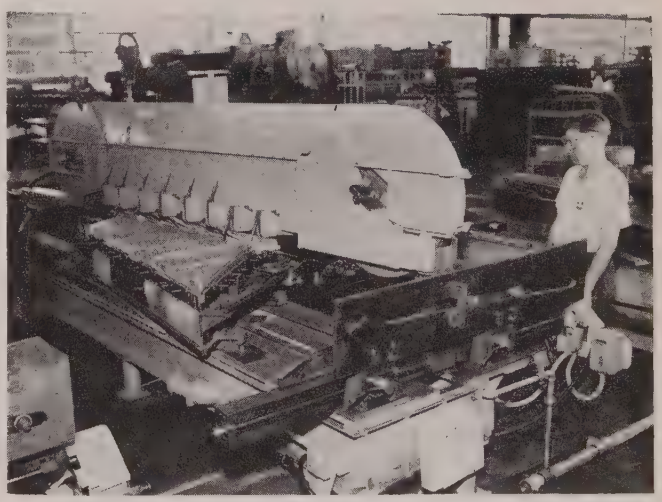

Fig. 9 – Contouring stabilizer honeycomb core

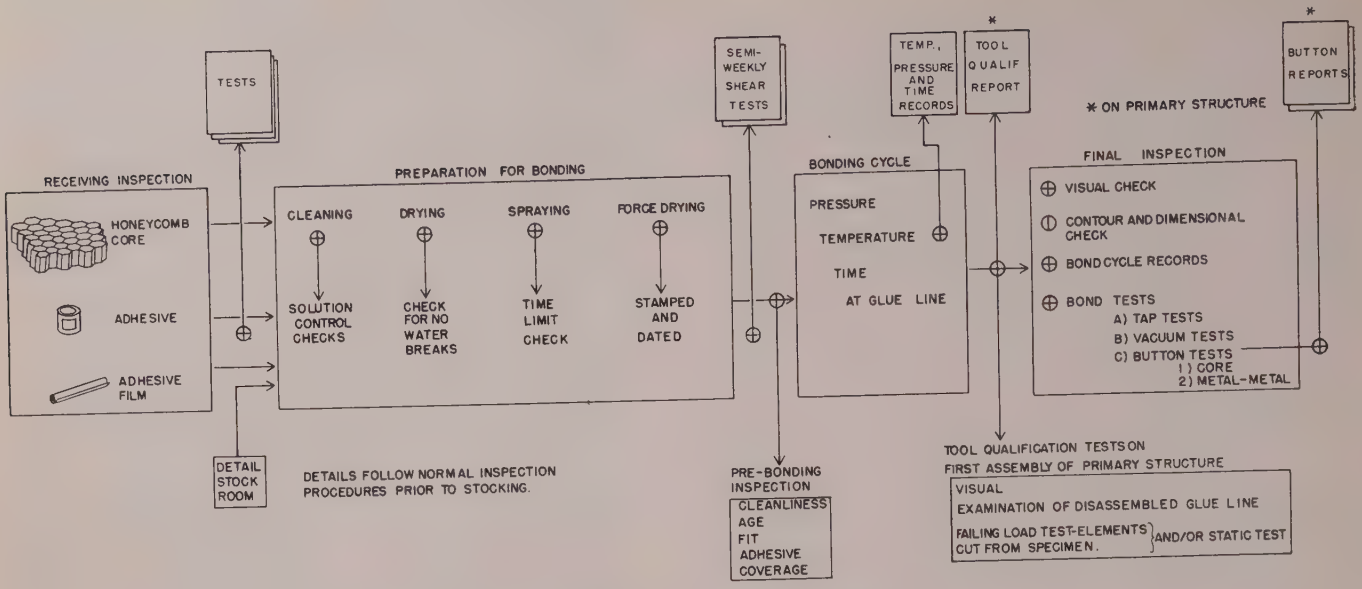

Fig. 10 – Process control for adhesive bonding

costs, complicates quality control, and reduces reliability. Layout of facilities should be such as to minimize handling.

3. Adhesive spraying should be done as soon as possible after hot-air drying of cleaned parts.

4. Production planning should be enlisted behind the effort to effect a "clean" operation; for example: parts should not be started on the bonding cycle unless they can be cleaned, dried, and sprayed within 12 hr. This is done easily enough by processing the parts through detail manufacturing in whatever job-lot sizes may be suitable, and then stocking them for bonding. The bonding supervisor can draw on this stock room according to the bonding load.

5. People who handle cleaned parts must wear white gloves – and clean gloves should always be available to workers. Perspiration left by the hands on a cleaned surface can account for as much as a 60% loss of bond strength, at the point in question. Cleanliness ideas can best be put across by means of continuing educational programs.

6. On the Matador project, new job classifications, such as "honeycomb assembler" and "Autoclave operator" were set up – so that seniority and union rules would not drain experienced personnel from the bonding group.

7. Since a greater number of less-experienced personnel was involved in the production than in all other phases of the efforts, it was found necessary to be explicit in presenting engineering information as to areas and thicknesses of adhesive application, cleaning processes, and handling procedures. More detailed data were placed on the engineering drawings and tool write-up cards. Information aids, such as a chart for converting bonding loads to pressure gage readings, were also made available to shop personnel.

8. Close liaison between engineering, tooling, and manufacturing was indispensable in the beginning, since it made possible the working out of detail problems as they arose and pointed the way to the solution of future problems.

E. The wave of quality control effort is still high on the Matador project. In addition to some very careful development work, under USAF contract, devoted to the search for new principles that might be applied to the inspection problem in connection with the final product, a great deal of on-the-spot test work has been accomplished by our company. The statistics obtained from this work have proved invaluable in effecting a workable operating inspection procedure within the present state of the art. Fundamentally, the reliability of the inspection procedures herein described is based on a careful step-by-step process control, plus certain verification checks at final inspection. The overall plan of control, as shown in Fig. 10, is based on the following factors (all of which contribute to the inherent reliability of the product and hence help to simplify the process control task):

1. The FM-47 adhesive has been chosen because of its excellent stability in storage. This greatly

simplifies the quality control problem by virtually eliminating the time element. The adhesive itself needs to be checked only once – as it comes into the plant.

2. Again, this adhesive was chosen to make the inspection practical, since it provides a good safety factor over the MIL specification shear strength of 2500 psi. FM-47 will average about 4000-psi tensile shear strength.

3. Adhesive film (FM-47) is used to reduce the criticalness of the core-to-skin bond, which obviates the need of obtaining 100% bond inspection.

4. Statistics has shown that the most important step is that of cleaning, drying, and coating, because once the coated part has been force-dried, it is relatively safe from contamination. In addition to the normal check of a chemical cleaning system, such as for solution control, for water break, and so on, this phase of the work is kept under surveillance by means of semiweekly shear tests of specimens processed along with production parts.

5. It should be noted that up to this step all parts are open to inspection, which points up the importance of the next step – inspection before bonding. Here, before closing up the article, the inspector verifies that all previous steps have been satisfactorily completed. He checks such things as fit, cleanliness, force-drying, and adhesive coating.

6. In the next step, which is the bonding cycle, the inspector simply requires evidence that the glue line has been subjected to the required temperature, pressure, and time – 300 F minimum for 30 min; 15 psi on core; 100 psi on metal to metal. This is made available by use of thermocouple and pressure gages, designed into the tooling and often recorded and controlled automatically. (See Fig. 11.)

7. The first part of any primary structure is turned over to the engineering materials laboratory for a tool qualification test. This is a destruction test because the part is disassembled so that the bond may be inspected and, where possible, elements are cut from the specimens at all critical points and static tested. The bond inspection covers such factors as adhesion, contact, cure, and porosity. This is done by laboratory people experienced

Fig. 11 – Matador wing tool and control panel

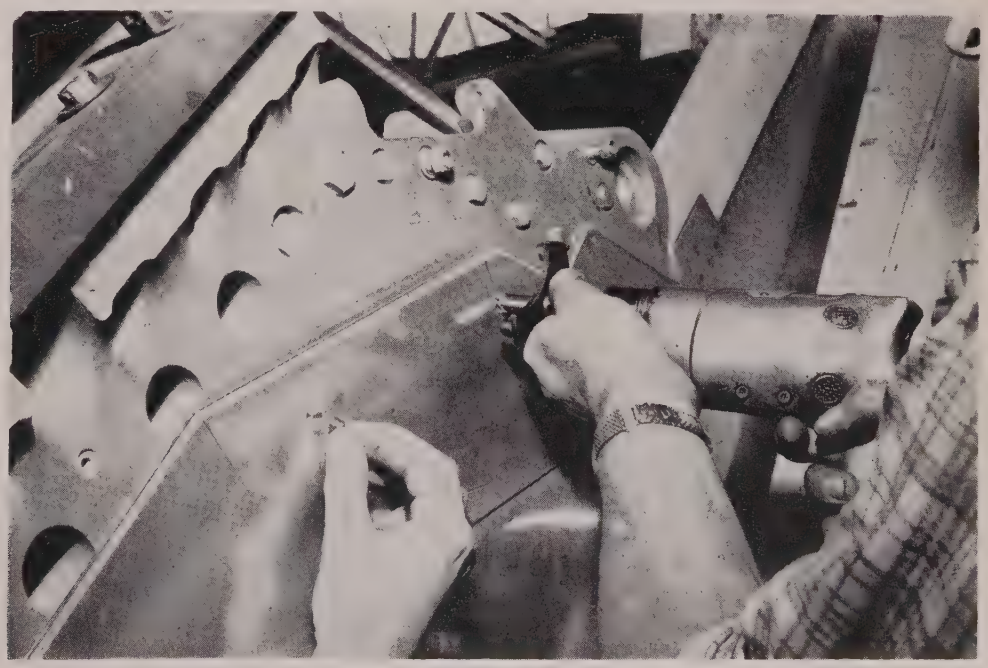

Fig. 12 – Removal of inspection button from metal-to-metal adhesive joint for examination of glue line

in this work. On the basis of evidence obtained in these tests, design and/or tooling corrections are made as required. Once the tool has been qualified, repetition of this test is generally not required. This test is repeated whenever there are significant changes to the design or tooling and on 1% of any primary structural components. Sometimes the tool qualification tests can be performed on the static test article.

8. All parts made after the tool qualification test are turned over to the quality control people, who see that all controls have been complied with, make visual, tap, and vacuum checks, and also button tests on skin-to-core and metal-to-metal bonds. Fig. 12 shows the removal of an inspection button from a metal-to-metal adhesive joint for examination of the glue line. Contact, cure, and porosity are checked. This type of inspection is employed in the case of primary structures at the location specified on the drawing by the structural design engineer. As production advances, demonstrating the consistency of a particular assembly, the button tests can be reduced. They are generally retained, however, because they provide specific information on the quality of the bond on the part being delivered, making it unnecessary to rely on the tool qualification test and the other indirect quality control tools. Many thousands of bond joints have been inspected, on the Matador project alone, for adhesion, contact, hardness, and porosity, in addition to a great number of static test check points of components and complete assemblies.

It may be concluded, therefore, that production adhesive bonding of flat and contoured primary structures, with all its inherent advantages, is today available to the industry. But it will continue to require the well-coordinated efforts of laboratory, engineering, tooling, manufacturing, and quality control to develop the necessary know-how to assure proper process control and to build the confidence that must exist on the part of all concerned in the excellence of the end product.

ORAL DISCUSSION

Reported by H. A. Helstrom,
Chance Vought Aircraft, Inc.

Question: What are the relative costs and ultimate life?

Mr. Burridge: In dollars per pound of surface, the cost is about half that of riveted construction. As far as ultimate life is concerned, considerable work has been done on fatigue characteristics. The indications are that it is at least comparable to riveted construction and will probably prove to be even better.

Question: What are the highest anticipated flight operating temperatures?

Mr. Burridge: The working temperatures are not excessive, for 130 F would be a fair maximum.

Question: How does the strength of the bond vary with temperature changes? Would 450 F for 5 min be acceptable?

Mr. Burridge: Fig. 4 of the paper plots our best data. Actually, some work has been done to improve the high-temperature creep problem. Ten minutes at 200 F would probably be satisfactory now, and by changing the curing temperature (from the present 300 F) the allowable shear strength could be improved. Five minutes at 450 F would be unsatisfactory with the present curing cycle.

Question: What has been your salvage experience if inspection testing shows the bond to be unsatisfactory?

Mr. Burridge: We usually take more button samples to determine exactly the nature of the salvage required. Standard salvages have been set up and made. Major castings and surface skins are sometimes removed and replaced. It is a major job and certainly not very easy, but we have had no complete scrappage yet.

Question: Why is a liquid adhesive used on the core and a film adhesive on the sheet?

Mr. Burridge: A liquid primer is used to apply light coats on both core material and facings to assure proper wetting of the adhesive surfaces to be bonded. The film may be attached to either core or faces according to convenience. It comprises the bulk of the adhesive weight.

Question: What experience have you had using this process in the construction of integral fuel tanks?

Mr. Burridge: We have had none, but perhaps North American might be a source to investigate.

Question: Are X-ray or ultrasonic inspections made?

Mr. Burridge: While Martin does not use these devices at present, we understand that GE has done some work with ultrasonic methods and that they have a thermal device under development for the Air Force.

Question: Does Martin use contoured and/or tapered sheet?

Mr. Burridge: No, at the present time we feel that the use of bonded doublers is less expensive.

Question: What tolerances are held in the fabrication of these parts?

Mr. Burridge: The cores are contoured to ±0.010 in. Maximum allowable mismatch at a joint to the casting is 0.015 in. Voids must be filled with adhesive, and the strength of the joint is probably not a function of thickness (at ordinary temperatures and up to 0.030 in. thick). Adding more adhesive tends to help solve the tolerance problem.

Question: What are the effects of moisture, fuels, and oils?

Mr. Burridge: Testing indicates fuels and oils do not affect the bond. Moisture is no problem except that the honeycomb cells fill up with water. Bonded joints show no loss in strength after immersion in water, oil, ethylene glycol, alcohol, or hydrocarbon fluid, or after exposure to salt spray.

Stress Corrosion of 12% Cr Stainless Steel

W. L. Badger, Thomson Laboratory, General Electric Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1954.

SEVERAL jet-engine manufacturers have, in the course of their experiences, encountered cases of stress corrosion cracking. This combination of stress and corrosive action results in individual cracks of a brittle, intergranular nature. These cracks concentrate the stress so that failure of the part may occur through further stress corrosion, fatigue, or tension because of reduced area.

A case of stress corrosion with which the author is familiar occurred in compressor rotor blades. These blades, which were made of 12% Cr, type 403 stainless iron, were found to be cracked along the leading edge, after a considerable amount of operation. The cracks (Fig. 1) occurred at random locations along the leading edge of the blade. As far as is known, they never extended into the blades more than is indicated in Fig. 1, and the cracks were always of an intergranular nature, as shown in Fig. 2.

The frequency of occurrence of cracks was re-

STRESS corrosion cracking is a combination of stress and corrosive action that results in individual cracks of a brittle, intergranular nature.

The author discusses such failures in compressor rotor blades made of a 12% Cr, type 403 stainless steel.

A laboratory technique was worked out for producing similar failures at will.

As a result of this study, it was recommended that compressor blades be stress relieved at 950 F. Since this has been done, no further cracking of the blades has been reported.

Tests with three alloys in addition to the type 403 showed the former alloys to be superior to 403 in regard to stress corrosion cracking.

■

The Author

W. L. BADGER has been in charge of metallurgical work in the Thomson Laboratory, General Electric Co., since 1941. Early in his career he transferred from chemical to metallurgical work and was closely associated with the late Sanford Moss in metallurgical problems involved in the development and production of the turbosupercharger. Mr. Badger is a graduate of Northeastern University and holds a degree in chemical engineering.

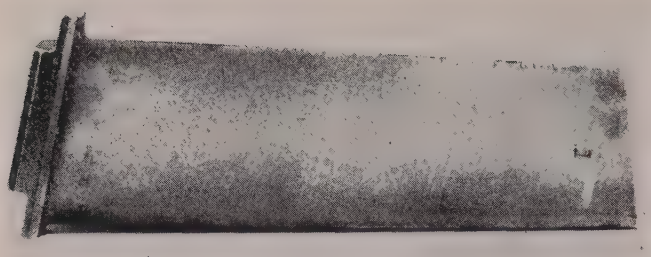

Fig. 1 – Cracked compressor blade

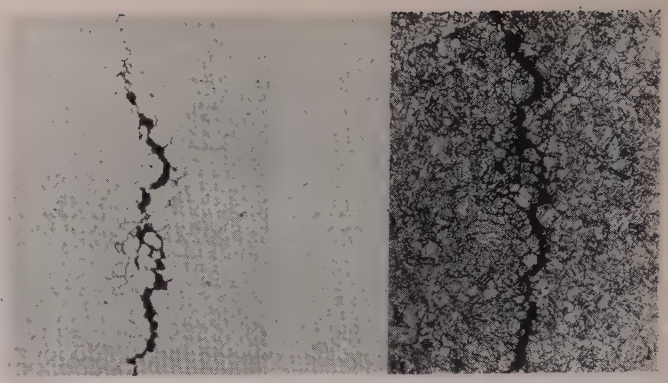

Fig. 2 – Cracks in compressor blade – left: unetched, right: etched (reduced from photomicrographs taken at 350 X)

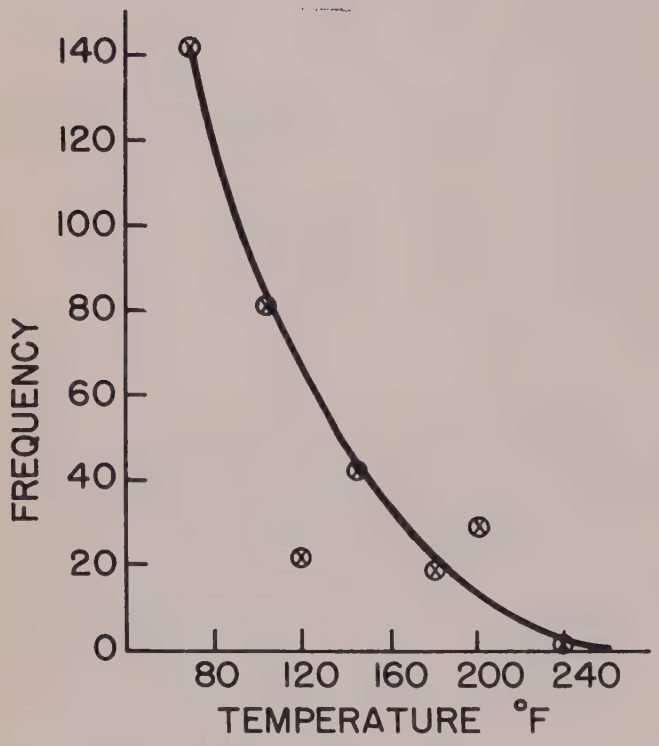

Fig. 3 – Frequency of cracking of compressor blades versus temperature

Fig. 4 – Compressor blade cracked in 1-1 HCl + 1% SeO_2

lated to the temperature of the various stages and, therefore, to the moisture available, as shown in Fig. 3.

A study was made to determine if similar failures could be produced quickly in the laboratory so that the cause and cure could be determined. It was found that one of the manufacturing operations involved hand polishing the leading edge on buffing wheels. In this operation more or less heat was generated by friction, depending upon the pressure applied. In some instances, this was sufficient to heat the blade edge into the transformation temperature range. As this material is air hardening this operation resulted in a hard blade edge of untempered martensite highly stressed in tension. A number of accelerated corrosion tests were made on blades treated in this manner. The several corrosive media tried would not produce cracks in short periods of time until H_2S or SeO_2 was introduced into the solution.

It was decided that the most reproducible test data were obtained using a corroding solution of 1-1 HCl + 1% SeO_2. With this corrosive medium, cracks identical to those occurring in the field (Fig. 4) could be produced in compressor blades at will in approximately 15 min of immersion.

The nature of the stresses was difficult to control and measure in the compressor blades, which had been locally heated on the edges. To obtain a more quantitative measure of the effect of stress and time on stress corrosion cracking, a series of tests was made using strips instead of blades. These strips were heat-treated and mechanically deflected to produce the desired stress. The fixture (Fig. 5) with the deflected strips was immersed in the corroding media and time to produce cracks determined.

These tests showed that a tensile stress of approximately 50,000 psi was required to cause cracking and that the data fall in a pattern similar to fatigue or endurance curves. (See Fig. 6.)

As this stress was several times the operational stress of the compressor blade and because of the random location of the cracks in compressor blades, it was determined that the residual stress in the leading edge was causing the trouble in the field.

Fig. 5 – Strips stressed for corrosion tests

This residual stress could be eliminated or reduced with a stress-relief anneal (which also tempered the martensite), as a final manufacturing operation. Corrosion tests were made on compressor blades with locally heated edges, which had been stress-relief annealed at several temperatures. The results of these tests are shown in Fig. 7.

On the basis of these tests, all compressor blades were stress relieved at 950 F. Since this time no further cracking of the blades has occurred.

During the course of this investigation, there has been considerable speculation as to the role of hydrogen in this cracking problem. No attempt was made here to determine the actual mechanism of stress corrosion cracking. However, the direct influence of the material hardness and the applied stress, together with the high stress-relieving temperatures required to lower the susceptibility, preclude the consideration that hydrogen alone caused the cracking.

As jet engines develop and efforts are made to reduce weight, the material requirements become more severe. The temperatures rise, necessitating the use of stronger alloys and different heat-treatments. Tests have been run using the same test procedures to determine the effect of hardness (tempering temperature) and alloy content on the stress corrosion resistance of several proposed alloys.

Three alloys, in addition to type 403 stainless, have been included in these stress corrosion tests. Their composition is shown in Table 1.

Each material was austenitized at its recommended temperature, quenched between steel plates

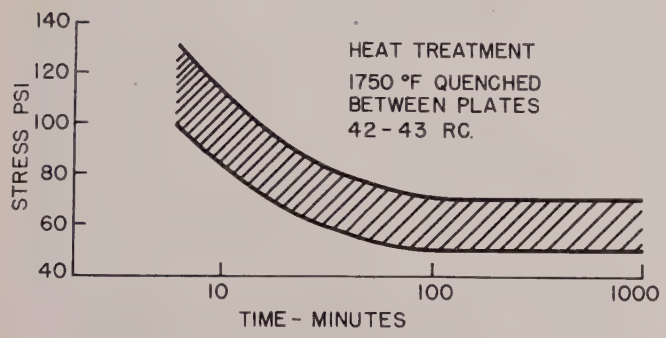

Fig. 6 – Stress versus time to crack for 12% Cr stainless steel in 1-1 HCl + 1% SeO_2

Table 1 – Composition of 12% Cr Alloys in Stress Corrosion Tests

Material	Composition (Nominal) C	Cr	Mo	W	V	Ni
403	0.10	12	—	—	—	—
12 Cr-Mo-V	0.30	12	3	—	0.25	—
12 Cr-W-Ni	0.12	12	—	3	—	2
12 Cr-Mo-W-V	0.20	12	1	1	0.25	—

Table 2 – Heat-Treatments and Life in Minutes in 1-1 HCl + SeO_2 at Several Stress Levels

Material	Condition: Quench	Temper	Hardness, R_c	Time to Failure at: 140,000 Psi	125,000 Psi	100,000 Psi	75,000 Psi
12% Cr B7B21 403	1750	None	42-43	—	7	10	20
	1800	350	38	70	80	150	450[a]
	1800	1000	33	20	40	85	225[a]
	1800	1050	28	45	70	300	No failure
	1800	1100	23	No failure in tests			
B5OR55 12% Cr-W-Ni	1850	350	51	25	30	50	300[a]
	1850	1000	47	300	375	575	No failure
	1850	1100	36	240	375	1000[a]	No failure
	1850	1200	30	No failure in tests			
B5OR311 12% Cr-Mo-V	2000	350	58	5	7	15	100[a]
	2000	1000	55	60	100	225	500
	2000	1200	36	250	550	No failure	No failure
	2000	1300	34	No failure in tests			
Crucible 422 12% Cr-Mo-W-V	1850	350	52	17	25	50	150[a]
	1850	1000	49	350	No failure	No failure	No failure
	1850	1100	39	No failure in tests			
	1850	1300	32	No failure in tests			

[a] Extrapolated.

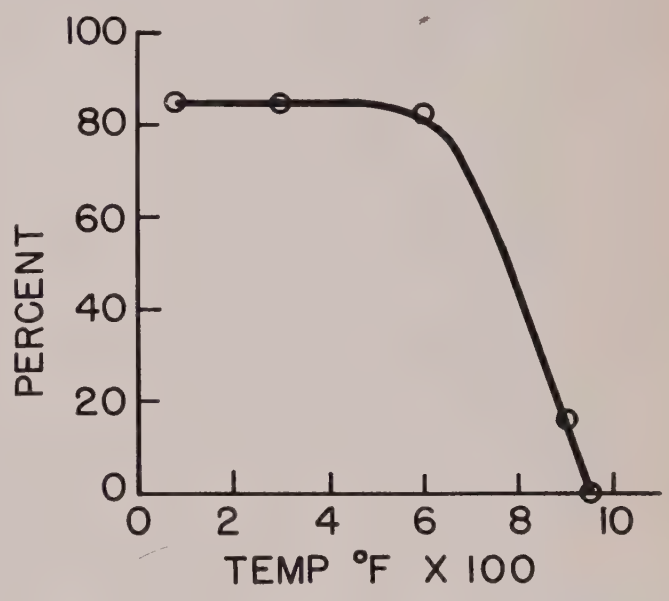

Fig. 7 – Percentage of cracks in corrosion test versus stress-relieving temperature

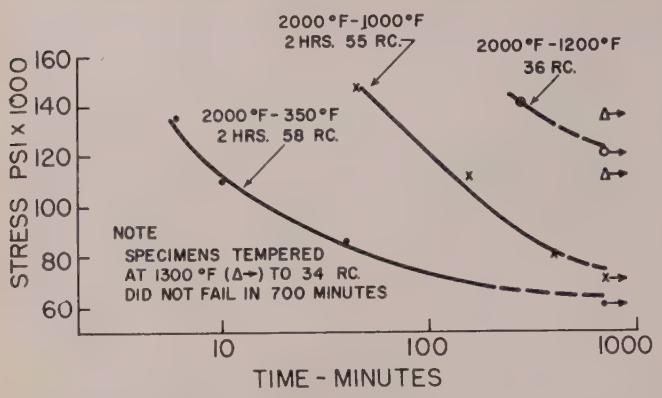

Fig. 8 – Stress versus time to crack in 1-1 HCl + 1% SeO_2 for 12 Cr-Mo-V

to keep the test strips flat, and tempered to get suitable hardness ranges. The heat-treatments and resulting hardnesses are included in Table 2.

The data obtained show typical curves for all alloys and hardnesses. Fig. 8 shows these curves for 12 Cr-Mo-V as an example of the data obtained. There are definite trends established, which show advantages for the alloy modified 12% Cr steels in that the time to failure at a given hardness and stress is much greater for these materials. Additional data are needed before definite effects can be determined. There is the further question as to the correlation of this test in 1-1 HCl + 1% SeO_2 with actual operational environments.

The life in minutes at several stress levels has been tabulated for each material and hardness level in Table 2. From this information, Fig. 9 has been plotted to show the life of each material versus the hardness. In this plot, it is apparent that the alloys show a definite superiority over the straight 12 Cr, type 403 steel for equivalent hardness in a stress corrosion test with 1-1 HCl + 1% SeO_2.

The author wishes to express his appreciation for the assistance of W. E. Jones, E. M. Lape, and F. L. VerSnyder in conducting these tests and preparing the discussion.

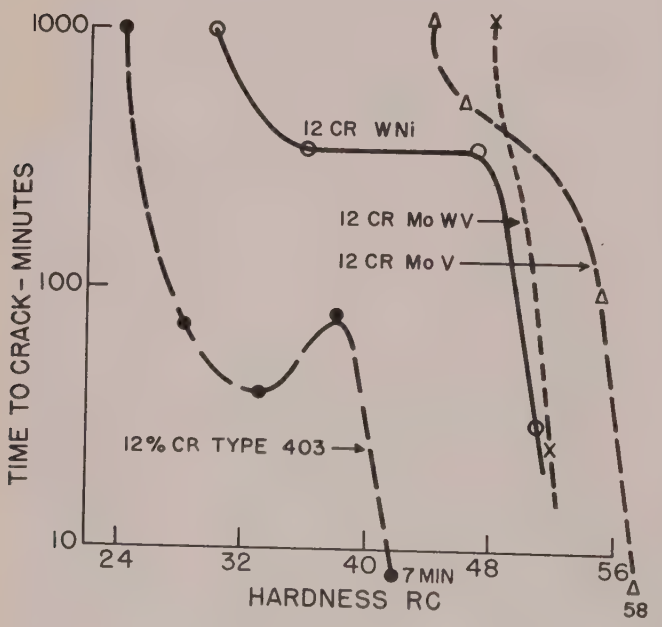

Fig. 9 – Time required to crack several alloys in 1-1 HCl + SeO_2 at 125,000 psi

DISCUSSION

Confirms Data for Type 403 Stainless

– Louis J. Danis
Eaton Mfg. Co.

WE have conducted tests similar to those described in the paper and can confirm that type 403 stainless developed its poorest properties when tempered at 1000 F to a Rockwell C hardness of 33-35. The tests also confirmed that a tempering temperature of 1000 F puts this material in the most susceptible condition for corrosion, whether or not residual stresses are present. This only confirms a large body of prior technical literature on this subject.

It has also been learned that stress-corrosion cracking can be minimized or eliminated by treating the compressor blade in such a way as to promote residual compressive stresses in the surface. One of the deburring methods, which involves plowing blades through a bed of slightly abrasive stones, can both smooth the surface and leave it in a condition of slight residual compression. There is an interesting possibility that the problem of stress corrosion can be even more safely controlled by adding such an operation to the strain-relieving treatment described by Mr. Badger.

In the discussion period Mr. Badger indicated that in actual practice strain-relieving temperatures in the level of 1100 F are used and specified. This is consistent with a hardness requirement of Rockwell C 20-26. We agree with the author that this is a safer hardness range to specify for type 403 than the R_c of 30-38 called for by the original specification of several years ago.

The author of this discussion wishes to express his appreciation to J. Sprague and William Cummins, who supplied some of the information used in preparing this discussion.

Feels Salt Spray Tests Are Nearer to Service Conditions

– John Preston
Wright Aeronautical Division

THERE are several questions regarding the data that appear to require further clarification:

(a) What were the heat-treatment and hardness of the compressor blades used in this investigation?

(b) Table 2 indicates a 100% improvement in the time to failure for specimens of type 403 that were stress relieved at 350 F in comparison to specimens tempered at 1000 F. Is there an explanation for the improved stress corrosion properties of the stress-relieved specimens?

(c) It was stated that stress relieving the blades at 950 F after final machining prevented further stress corro-

sion cracking. Did specimens similarly stress relieved indicate a longer time to failure?

(d) Fig. 9 indicates an inflection point for type 403, where a specimen at 32 R_c showed a lesser time to crack than a similar specimen at 38 R_c. Does the author have any comment on this behavior?

The author questions a correlation between stress corrosion testing in an acid and actual operational environments. We are conducting stress corrosion tests of the type 403 stainless steel in a 20% salt spray atmosphere, which approximates service conditions better than an acid environment. This type of test has the advantage of long exposure periods in comparison to the short times for specimens submerged in an acid.

The test consists of loading a beam-type specimen[a] to obtain a stress below the elastic limit using an SR-4 strain gage (type A-7), as indicated in Fig. A. The fixture, with the stressed specimen, is then coated with wax, and the top surface, which is in tension, is thoroughly cleaned. A specimen ready for exposure in the salt spray atmosphere is shown in Fig. B. The specimen is examined visually for cracks at the end of each 24-hr cycle. If the specimen has not cracked the corrosive products are removed from the exposed surface using a brush and pumice, and testing is continued.

The data for beam-type specimens were plotted on a log-log scale (Fig. C). It can be seen that there is a straight-line logarithmic relationship between time to failure and applied fiber stress, provided heat-treatment and hardness are constant. The specimens were austenitized for ½ hr at 1800 F, aircooled, and then tempered for approximately 2 hr at 1000 F, to achieve a hardness covering the range of 32/38 R_c. The specimens were obtained from the same heat of material. Outer fiber stresses between 30,000 and 60,000 psi were applied to the specimens.

There was a 300% improvement in the time to failure for specimens of 34 R_c in comparison to those of 38 R_c. Specimens failed with an applied outer fiber stress of 30,000 psi. All stress corrosion failures in the salt spray atmosphere have been intergranular, and these are similar to the type of failures encountered when specimens are exposed in an acid.

The improved stress corrosion properties for the lower-hardness type 403 material is believed to be due to the amount of grain boundary energy. The energy is affected by chemical composition, heat-treatment, and manufacturing operations. It is probable that the grain boundary energy influencing the susceptibility of the type 403 to intergranular stress corrosion is primarily affected by the type of carbide precipitation associated with tempering temperature. We have not been able to determine the amount of grain boundary energy to prevent stress corrosion cracking. However, we believe that this theory has more significance after examining the data of the author.

The author investigated four martensitic stainless steels having 12% chromium and added percentages of other alloying elements. The alloys were tempered at 1000 F and above to procure a hardness in the range of 32/38 R_c. The data indicate that improved stress corrosion properties are associated with increasing tempering temperature and not hardness, provided the range of 32/38 R_c is maintained. Therefore, it is believed that the improved properties are related to the grain boundary energy, which was affected by chemical composition and heat-treatment, since the specimens were similarly machined. The modified 12% chromium steels permitted the use of higher tempering temperatures to achieve a hardness of 32/38 R_c, and these alloys exhibited better stress corrosion properties than the type 403.

The stress corrosion tests in an acid and salt spray are

Fig. A – Applying a stress to beam-type specimen

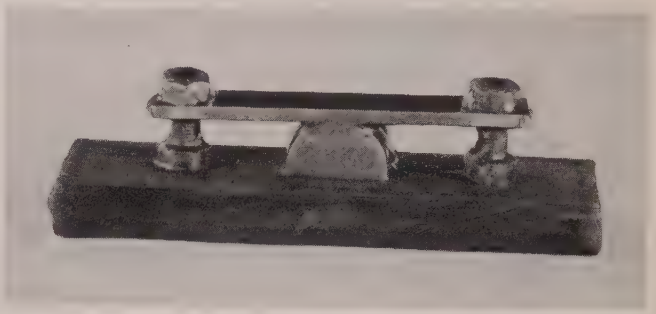

Fig. B – Beam-type specimen prepared for exposure in 20% salt spray environment

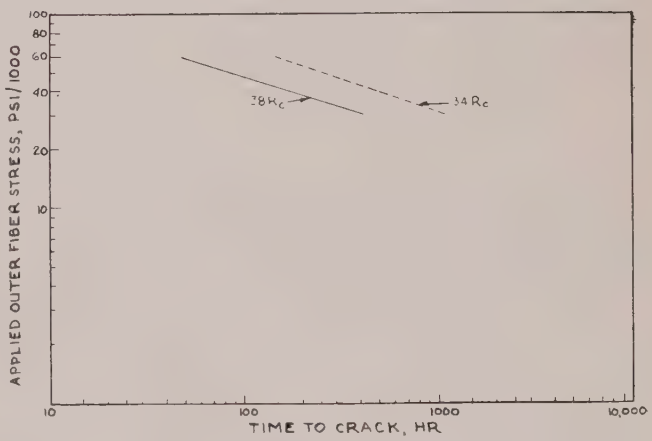

Fig. C – Stress corrosion properties of type 403 steel in 20% salt spray environment. Beam-type specimens. Heat-treatment: austenitized for ½ hr at 1800 F; aircooled; and then tempered for 2 hr at 1000 F

[a] Discussed in "Symposium on Stress Corrosion Cracking," pp. 395-410: "Some Observations of Stress Corrosion Cracking in Austenitic Stainless Alloys," by M. A. Scheil. Pub. by ASTM-AIME, Philadelphia and New York, 1945.

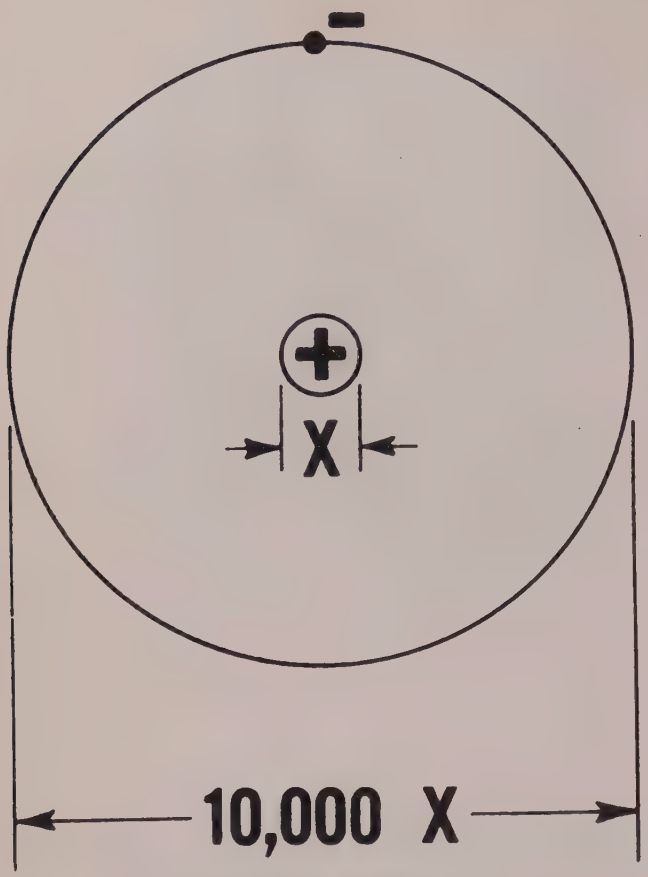

Fig. D – Diagram of hydrogen atom

similar. A significant difference between them is that failures can occur at much lower stresses for specimens exposed in a 20% salt spray atmosphere than those submerged in an HCl + SeO_2 solution. It is felt that type 403 steel will crack, due to stress corrosion, in a salt spray environment at stresses below 30,000 psi, provided the material has been tempered at 1000 F, but the fracture times would be very long. Therefore, it is believed that a log-log relationship, similar to a stress rupture curve, is an improved form for representing stress corrosion data obtained from specimens exposed in a salt spray atmosphere.

Nuclear Mechanics Shows How Hydrogen Helps Cause Cracking

– E. T. Neubauer
Trane Co.

MR. Badger made the following remark:

"During the course of this investigation, there has been considerable speculation as to the role of hydrogen in this cracking problem. No attempt was made here to determine the actual mechanism of stress-corrosion cracking. However, the direct influence of the material hardness and the applied stress, together with the high stress-relieving temperatures required to lower the susceptibility, preclude the consideration that hydrogen alone caused the cracking."

This is technically true because hydrogen **alone** did not cause the trouble. His information proves that the stress condition was also a factor. However, there is a strong possibility that hydrogen did play an important part in this problem when the triaxial tensile stress of the hydrogen is added to the residual stress. The purpose of these remarks is to show, by nuclear mechanics, how hydrogen can play a part.

During the past two decades the physicist has made tremendous strides in gaining knowledge of atomic structure and nuclear mechanics, to the extent that the atomic bomb was made possible. Some of this knowledge has found its way into other fields, and we are assured that it is useful and will lead to a better understanding of some of these problems.

In order to show how it is possible for hydrogen to play a part in this stress corrosion, it is necessary to have a mental picture of the hydrogen atom. Fig. A shows a diagrammatic sketch of this atom and illustrates the fact that the nucleus is only 10^{-4} times (one ten-thousandth) the size of the whole atom. In spite of this small size, 99.95% of the weight is concentrated in this nucleus. The volume ratio is, therefore, 10^{12} to 1 or a trillion to 1.

It is a known fact that hydrogen embrittlement or stress corrosion, such as was found in the tests conducted by Mr. Badger, occurs when the stressed strips were immersed in a solution of hydrochloric acid. This phenomenon occurs only when "nascent" (ionized) hydrogen is present. An ionized atom is an atom with at least one electron removed. The hydrogen atom has only one electron, therefore, when it is ionized it has no electrons and only the tiny nucleus remains. This tiny nucleus is small enough to enter the intermolecular spaces of any material. If it picks up a free electron (electrons are free in all space) when it is in the steel, it again goes back to its normal size, creating hydrogen gas pressure that is triaxial tension. If this occurs in a fine-grain, hard steel, sufficient hydrogen will be retained to cause an appreciable stress.

It is to be noted that stress corrosion occurs only where the residual stress is tension. This residual tensile stress added to the tensile stress caused by the hydrogen results in failure of the steel.

Variable conditions will be noted from stage to stage of the compressor because of two variable factors: (1) The temperature and pressure of the air and moisture vary. (2) The temperature of the blades (material) varies. Both factors affect the resultant hydrogen stress.

If it were possible to retain the hydrogen it would, by this phenomenon, be capable of pumping a pressure of 1,200,000,000 psi, which, according to Dr. Gamow, is the crushing strength of the atom. Obviously no material is capable of holding such pressure. On the other hand, all materials are not destroyed when exposed to the hydrogen ion, as the hydrogen gas can readily escape from soft steels or from coarse-grained materials. It can only be retained in close-grained, hard steels, and even then the molecular activity and spacing are such that a temperature rise of only 250 to 300 F is required to increase the molecular activity of the steel to the point where the hydrogen readily escapes.

ANALOGY

Picture a large pile of boulders each the size of an ox. If one had oxen outside of the boulders who could change size (without mass change) so as to become as small as fleas, they could then easily enter the rock pile by going through the open spaces. Once inside, and changing back to their original size, they would then cause a tremendous internal stress.

Other Data Agree With Authors' Results

– W. S. Coleman, Jr., and M. Simpson
GMC Research Laboratories Division

OUR work done on determining a stress-relieving temperature of heat-treated type 403 stainless steel is in good

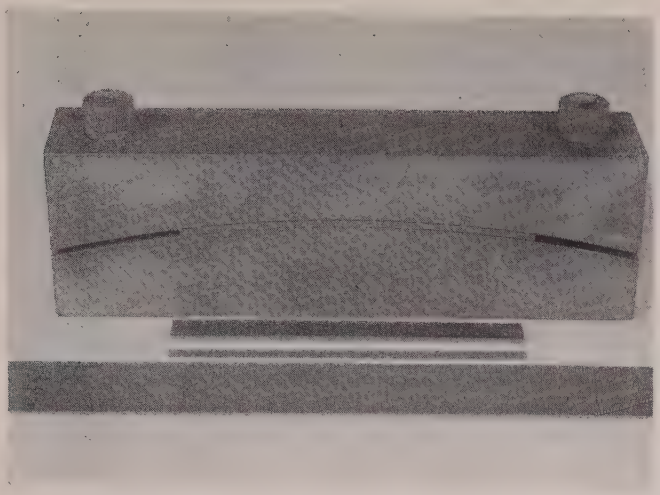

Fig. E – Curved form and specimens used for stress-relief temperature tests on type 403 stainless steel

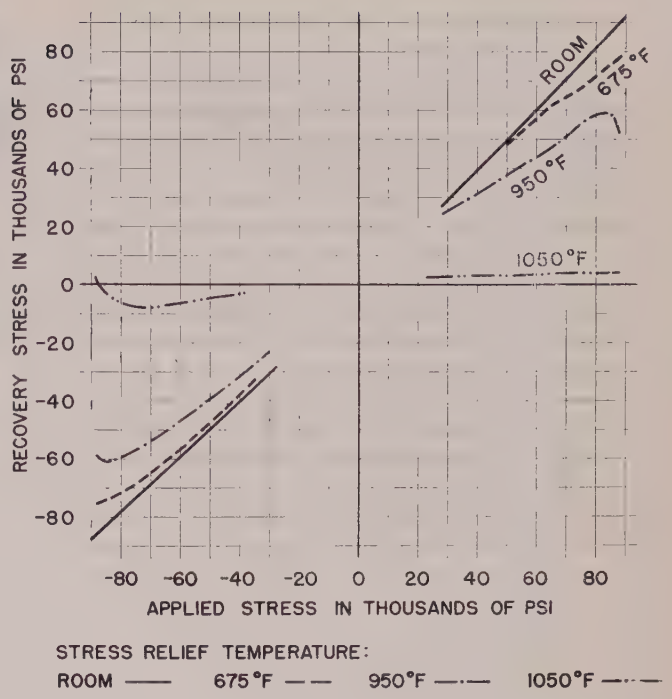

Fig. F – Recovery stress versus applied stress of type 403 stainless steel held at various stress-relieving temperatures for 2 hr

agreement with that of Mr. Badger. The approach to the problem was different in that we were interested in learning to what degree a stress-relieving heat-treatment is effective, and his principal concern was finding a particular treatment that would reduce the residual stresses in the blades to a nonharmful level.

The type 403 (turbine quality – 410) stainless steel used in our tests was heat-treated as follows:

1. Flatten between plates in furnace at 1500 F.
2. Heat to 1780 F for 20 min.
3. Aircook.
4. Temper at 980 F for 1 hr.
5. Resultant hardness 30-38 R_c.

A series of 0.06 x 0.47 x 3.00-in. flat strips of the subject material was placed in the curved form shown in Fig. A, heated for 2 hr to various stress-relieving temperatures, released from the form, and then analyzed for residual stresses. The residual stresses were measured by carefully grinding successive layers of metal from the test strips and noting the curvature changes. By taking into account the free curvature of the specimen after release from the form and the results of the residual stress analysis, we found that it was possible to calculate the stresses in the specimen after stress-relief treatment while still in the form.

The results of the analyses for the particular temperatures tried are shown in Fig. B. "Applied stress" is that imposed by the curved form previous to heating. The line marked "Room" is not exactly a straight 45-deg line since small residual stresses were found in the specimens as processed. "Recovery stress" is that apparent in the specimen after heating while yet restrained by the form. For example, the chart shows that for an applied tensile stress of 70,000 psi at room temperature, a recovery stress of 70,000 psi would result. However, if the temperature of the specimen under stress was raised to 675 F, a recovery stress of 65,000 psi would result. Similarly, if the temperature was raised to 950 F, the recovery stress would be 50,000 psi, and at 1050 F the recovery stress would be, for all practical purposes, zero. An examination of the data shows that stresses were relieved in varying amounts over all levels investigated. Note that a vertical line drawn through the curve family at any point of applied stress and plotted on the basis of recovery stress versus stress-relieving temperature would be very similar to that shown in Fig. 7 of Mr. Badger's paper. At the higher stress levels for the specific annealing temperature of 950 F, approximately 35% of the induced stress is relieved. Our data show further that, for nearly complete elimination of residual stresses, a minimum relieving temperature of 1050 F must be used for this material.

The investigation of the author shows that applied tensile stresses of 50,000 psi or below do not cause stress-corrosion cracks. He also shows that, in the actual blades (which were locally heated to produce residual stresses and then stress relieved at various temperatures), the cracking was eliminated when the stress-relief temperature reached 950 F. This does not necessarily mean that the residual stresses were eliminated but rather that they were reduced to or below a level of 50,000 psi, which agrees well with the data from our investigation.

Author's Closure To Preston Discussion

In regard to (a), I believe the blades were austenitized at 1800 F and tempered for 2 hr at 1050-1100 F to achieve a hardness of 32-38 R.

Concerning (b), I do not have any explanation for the improved results for specimens stress relieved at 350 F in comparison to those tempered at 1000 F.

Concerning (c), The effect of stress relieving the specimens at 950 F after machining on the stress corrosion properties in an acid has not been investigated.

Concerning (d), I do not have an explanation for the inflection point in Fig. 9 for the type 403 material.

Locomotive Power and Performance

With Special

THE small amount of power employed to move large masses on our railroads is seldom understood by men primarily interested in highway and air transportation.

Locomotives, however, have definite hauling limitations, which are controlled by the actual weight that is placed on the driving wheels.

With standardization of unit size and weight, the number of units and the aggregate horsepower of locomotives required to operate our railroads is surprisingly small.

The transmission rating and flexibility of electric,

THE inherent simplicity of the gas turbine and its well-known success in aircraft applications is leading to its consideration for locomotive use.

As a matter of fact, gas turbine locomotives have already found limited use by a few railroads throughout the world. The author discusses these applications and some of the lessons learned from them.

He points out that, although the first gas turbine locomotive to be put in service was built in 1941 – the same year that the first commercial diesel locomotive was placed in service – the latter has forged rapidly ahead, so that today the railroads are about 75% dieselized.

What, then, has held the gas turbine locomotive back? Mr. McGee points out that two of the most significant factors responsible are:

1. Metallurgical problems – the need for materials capable of withstanding the high temperatures encountered.

2. High fuel consumption.

He discusses both these factors – what has been done and what, in general, is still to be done – to solve them.

The Author

P. A. McGEE is currently Transportation Consulting Engineer in New York. He attended the London School of Science and Technology and graduated in 1909. Mr. McGee has worked with railways in London; Melbourne, Australia; South America; and the United States. He was also employed by the General Motors Corp. in connection with application and sale of diesel locomotives.

Requirements

Reference to Gas Turbine Locomotives

P. A. McGee, Consulting Engineer

This paper was presented at the SAE National Diesel-Engine Meeting, Chicago, Nov. 3, 1953

diesel, and gas turbine locomotives is a very important factor in converting powerplant horsepower to the tractive effort.

The fuel consumption index is a true measure of motive power efficiency and thermal economy.

These outstanding features of our railroad system will be considered in relation to the most modern type of diesel, electric, and gas turbine locomotives.

With this background, we will discuss the popular appeal, general design, and performance record of gas turbine locomotives in revenue service.

Five Transportation Industries

Up to the middle Thirties, the members of the SAE were primarily interested in highway and air transportation. Today, our railroads are approximately 75% dieselized, and the preponderance of marine power development, especially in Europe, is largely diesel. As a result, the members of the SAE are now interested in the basic power requirements of our five principal methods of transportation, namely:

Highway . . . Private automobile, bus, truck.

Railroad . . . Passenger, freight, and switching services.

Aviation . . . All types, civil and military.

Marine . . . High seas, coastal and inland waterways.

Pipe Line . . . Oil, gas.

There is a large difference in the application of an engine to the various methods of transportation. The resistance to motion of the various methods is best explained by considering the normal horsepower per ton applied to each of the industries.

Table 1 gives a rough approximation of the generally applied power.

The exact horsepower per ton will vary slightly with the particular conditions encountered. This variation occurs principally with rail and highway transportation.

It is not unusual for a railroad to haul a high-speed, 4500-ton freight train over a rolling profile with a 4500-hp diesel locomotive and then add a helper locomotive that may vary from 3000 to 6400 hp over a ruling grade.

Railroad Motive Power Problem

The comparatively small horsepower required per ton with normal railroad transportation, especially in freight service, will be noted in Table 1. The steel tires on the locomotive and trailing cars running on the steel rails give a minimum of rolling resistance. With long trains, the effect of head-end wind resistance on the train mass is a minimum per ton. As a result, long passenger trains may have a tractive resistance per ton less than 5% of that of an automobile. With heavily loaded freight trains, the railroad advantage over highway transportation may be even greater.

This basic condition practically insures the continued use of our railroads for mass transportation,

provided arbitrary rules and regulations do not interfere with the natural and inherent advantage.

Rail transportation, however, has one decided power disadvantage, compared with highway transport. Railroad locomotives must generally limit their starting tractive effort to 25% of the weight on their driving wheels. Rubber tires to dry asphalt or concrete can develop tractive effort in excess of 50% of weight on drivers.

In normal running, a railroad should not deliver a tractive effort in excess of 18% on its driving wheels so as to control slipping and wheel spinning at high speeds. These adhesion limitations control the weight of the train, which can be started and hauled over any given grade by a locomotive. In the United States, railroad grades seldom exceed 2%, whereas an automobile or truck can take a 10% grade in its stride.

A locomotive should, therefore, be able to deliver a tractive effort of 25% of its weight on driving wheels for a short period to start a train, and it should be able, in freight service, to deliver a tractive effort of 18% of its weight on drivers continuously or at least for a period sufficient to cover the ruling grade.

These limitations are very important and have lead to the development of generally accepted standards in axle loads. wheel diameters and loads, transmission capacity, and basic horsepower ratings per diesel locomotive unit best suited to the permissible axle loads, clearances, and general operating pattern of all Class I railroads in the United States.

Therefore, instead of the individual steam locomotive patterned to suit the particular axle loads, clearances, and facilities of individual railroads, we have today standard diesel units that are free to run on practically all Class I railroads from the Atlantic to the Pacific. The operating, fueling, inspecting, and indeed the maintenance of these units can be carried out without any special instructions on practically all railroads today.

With these standards established, all locomotive manufacturers are free to mass-produce comparable units, and the railroads have, in consequence, benefited not only in the initial cost of the units, but in their flexibility with increased usage and availability. Standardization has led to the use of a smaller number of locomotives than would be required if locomotives were tailored for individual railroads, as was the practice with steam power.

Locomotives Employed in Railroad Operation

The number of locomotives required to operate our Class I railroads and their required horsepower rating may be easily estimated from the diesel units that now supply about 75% of our railroad transportation.

Table 2 shows the number and horsepower of diesel units in service on Class I railroads on Jan. 1, 1953. For general convenience, these units are grouped under three classes of traffic, namely, passenger, freight, and switching. Actually, some of the units shown under freight and switching operate in passenger and other services. This diversity of service is becoming more pronounced as the application of diesel power progresses.

The point of greatest interest brought out in Table 2 is, I believe, the small number of locomotive units and the comparatively small horsepower required to operate about 75% of our extensive 223,000 route-miles of Class I railroads. Although the railroads had as many as 64,800 locomotives in 1925, the indications are that approximately 32,000 modern units, with an average weight of 125 tons, developing about forty-two million horsepower, would cover the total peak traffic of Class I railroads in the war years.

The comparatively small inventory of locomotive units is explained by the efficient use of railroad power and the high transportation output per hour developed per locomotive unit.

With a representative number of railroads. I find that the following average use through a 12-month period is obtained with diesel locomotives:

In passenger service, average miles per locomotive month, 17,100.

In freight service, average miles per locomotive month, 8,320.

In switching service, average hours per locomotive month, 520.

With the same railroads, the average gross trail-

Table 1 – Approximate Horsepower Required per Ton with Various Transportation Industries

		Hp per Ton
Highway:	Private Automobile	50 to 100
	Bus	15 to 30
	Truck	15 to 25
Railroad:	Passenger Mainline	3 to 5
	Passenger Rail Car	5 to 8
	Freight, High Speed	1 to 1.5
	Freight, Slow Speed	85 to 1.0
Aviation		100 to 250
Marine:	High Seas	0.25 to 1.0
Pipeline		Less than 1

The normal horsepower ratios given are for the usual speeds encountered with the types of services indicated.

Table 2 – Standard Diesel Locomotive Units in Service on Jan. 1, 1953

Type of Service	Wheel Arrangement per Unit	Average Weight per Unit, Tons	Hp per Unit	Number of Units in Service, 1/1/53
Passenger	A1A-A1A	155	1800 to 2400	1,834
Freight and general purpose	B-B	120	1350 to 2400	12,292
Switching	B-B	100	1200 & less	7,914
Total				22,040[a]
Total Horsepower—All Units:				29,684,892

In addition to the above, there were approximately 16,450 steam locomotives, 756 electric locomotives, and 20 of other types in service at the end of 1952.

[a] Simmons-Boardman survey of 1953 diesel-electric units in railway service.

ing ton-miles per unit train-hour in freight service is over 25,000. With a 3-unit locomotive, this gives 75,000 gross trailing ton-miles per train-or locomotive-hour. This is the actual average performance for the railroads considered. Not many years ago, 20,000 gross trailing ton-miles per train-hour was considered a satisfactory operation.

Analyzing further the horsepower capacities of the diesel units, shown in Table 1, we note that the units shown in freight service averaging 1500 hp account for nearly 56% of all the units produced. This value is significant and indicates that locomotive multiples of 1500-hp units, which give locomotives of 1500 hp, 3000 hp, 4500 hp, or 6000 hp, best suit the general requirements of our American railroads.

The steam, electric, and gas turbine locomotives employed on our Class I railroads do not comply with these standards and in consequence would be limited in their general application, as we shall show hereafter.

Power Transmission

The one outstanding advantage of steam reciprocating locomotives was the fact that the engine could be directly coupled to the driving wheels.

With all internal-combustion, automotive powerplants, a torque converter of some kind is required between the powerplant and the driving wheels. With highway equipment, gearshifts or hydraulic converters are generally employed. In railroad service, this country has standardized on electric transmission with individual axle drives for locomotives.

The standard electric motors employed are the transmission part of the simplest and most reliable electric locomotive ever developed. These motors may be supplied with electric power through a resistor, motor-generator set, mercury rectifier, or ignitron to give a so-called straight electric locomotive. The motors may be supplied with electric power from a generator driven by a diesel engine or gas turbine or, if you like, a steam turbine.

In all cases except the straight d-c electric locomotive, the torque converting motors may be similar for comparable service, and are directly comparable on a cost basis.

This transmission is heavy and, if it were not standardized and produced in quantity, it would be very expensive. It is, however, an ideal transmission in its functioning and its weight, at least with freight locomotives, can be profitably employed to develop tractive effort.

The transmission should comply with the requirements outlined when discussing the motive power problem. It should be capable of delivering a tractive effort equal to 25% of the weight on the driving wheels and deliver continuously up to 18% of the weight on the drivers in freight service. Furthermore, the weight on the driving wheels should be a maximum to deliver the normal horsepower rating of the powerplant at minimum acceptable speed.

There is a great temptation in the design of electric locomotives and even with diesel and gas turbine locomotives to exploit the low weight of the powerplant to the exclusion of useful adhesion weight for traction. This defect is largely the result of attempting to apply a-c motors with poor torque characteristics to electric locomotives.

In the final analysis of a locomotive transmission rating, the ratio between the permissible maximum and minimum speeds explains the capacity and flexibility of the unit. This ratio gives the range of speed over which the full horsepower of the locomotive can be employed continuously.

The simplest way to explain this question of transmission capacity and flexibility is to consider typical modern gas turbine, diesel, and electric locomotives, as applied to American railroads. Such a comparison is shown in Table 3. We will later refer to the actual tractive effort curves of the locomotives shown in Table 3.

Table 3 refers in column 1 to the Union Pacific gas turbine locomotive, the four diesel freight locomotives that may be developed from a standard 1600-hp unit, the straight electric locomotive in 2500-hp and 5000-hp sizes, and the ignitron electric locomotive in 3000-hp and 6000-hp combinations as recently delivered to the Pennsylvania Railroad.

In succeeding columns, we show the horsepower per unit, the wheel arrangement, the weight per axle, the total weight, the tractive effort, and the speed at continuous tractive effort, the maximum permissible speed, and finally the important ratio

Table 3 – Modern Gas Turbine, Diesel, and Electric Locomotives Locomotive Horsepower, Wheel Arrangement, Weight, and Tractive Effort Characteristics

	Hp per Unit	Wheel Arrangement	Weight, lb/1000		Continuous Rating		Maximum Permissible Speed, mph	Ratio of Maximum Speed to Continuous Speed
			Per Axle	Total	Tractive Effort, lb	Speed, mph		
Gas Turbine Locomotive	4500[a]	B-B-B-B	68.8	550	105,000	12.9	65	5.0
Diesel Locomotive	1600[a]	B-B	62.5	250	52,500	9.3	65	7.0
	3200	2 (B-B)	62.5	500	105,000	9.3	65	7.0
	4800	3 (B-B)	62.5	750	157,500	9.3	65	7.0
	6400	4 (B-B)	62.5	1000	210,000	9.3	65	7.0
Electric A-C Motor Locomotive	2500[b]	B-B	60.0	240	35,400	26.5	65	2.1
Electric A-C Motor Locomotive	5000	2 (B-B)	60.0	480	70,800	26.5	65	2.1
Electric D-C Motor Locomotive	3000[b]	B-B-B	62.50	375	66,000	17	63	3.7
Electric D-C Motor Locomotive	6000	2 (B-B-B)	62.50	750	132,000	17	63	3.7

[a] Gas turbine and diesel locomotive hp rating is input to traction generator.
[b] Electric locomotive hp rating is at the driving wheels.
Tonnage ratings are normally proportional to weight on drivers.

of maximum speed to permissible continuous speed.

Consider, first, the characteristics of the standard diesel locomotives. They have a weight of 156 lb per hp and could be designed to a weight of less than 130 lb per hp. Their continuous rating is nearly 21% of the weight on drivers, so these units, even at 156 lb per hp, are practically foolproof. The ratio of their maximum speed to continuous speed is 7, which gives a very wide range in operation.

The Union Pacific gas turbine locomotive has a weight of 122 lb per hp and a correspondingly low continuous rating of 105,000 lb at approximately 19% adhesion as against a continuous tractive effort of 157,500 lb, with a comparable 4800-hp diesel locomotive. The ratio of maximum speed to continuous speed is 5. The difference between the 4500-hp gas turbine and the diesel 4800-hp locomotive is that the diesel employs 50% more transmission capacity and, with equal axle loads, could haul 50% more tonnage on a given grade.

The electric locomotives have continuous tractive efforts of 14.7% and 17.6% of weight on driving wheels, respectively. They have fully exploited the inherent weight efficiency of their equipment with, however, a great sacrifice in flexibility. Their ratio of maximum speed to continuous speed is 2.5 and 3.7, respectively. These ratios are lower than those generally obtained with diesel passenger power. The standard diesel locomotive of 3200 hp or 6000 hp has 40 to 50% more transmission capacity than the corresponding electric units. The standard diesel can, in consequence, haul over 40 to 50% more tonnage on a given grade. The favorable weight ratio of the electric units is, therefore, obtained at a great sacrifice of hauling capacity.

Fuel Consumption on Class I Railroads

This most important index of the efficiency of motive power could well lead to a lengthy discussion involving the thermodynamics of the different types of motive power. Such a study would be of very little practical value because of the variable and generally low load factor with locomotives in railroad service.

Naturally, the best measure of fuel or energy requirements is the values obtained in actual everyday operation. In this connection, the ICC publishes fuel statistics regularly. I have shown in Table 4 the unit fuel consumption values as published by the ICC for the year 1952 for the four principal sources of energy applied to locomotives. There are no official published figures relating to the gas turbine locomotive, and we cannot include values for this method of operation without first discussing the gas turbine locomotive in detail, which we will do later.

It is not necessary to consider all the fuel and traffic statistics of our railroads for 1952, but I have added in Table 4 fuel cost values that are of interest. In the lower part of Table 4, I have shown the total approximate expenditures for transportation fuel in 1952. Following that value, I have shown the approximate cost of fuel for 1952 on the assumption that all operations would be carried out as follows:

By steam locomotives employing coal at $5.50 per ton.

By steam locomotives employing bunker C fuel oil at 4.5¢ per gal.

By diesel locomotives employing distillate at 9.8¢ per gal.

By electric locomotives employing power delivered at 10.52 mills per kw-hr and employing distillate fuel for car heating in passenger service.

The saving with complete diesel operation over that of steam-oil is over half a billion dollars per year and over steam-coal is approximately $450,000,000 per year. With the normal combination of coal- and oil-steam power prevailing over the last 10 years, the fuel saving with complete diesel operation, as against steam power, would exceed one-half billion dollars.

In the case of complete electric operation at the power rates prevailing in 1952, the saving with diesel operation would be practically the same as

Table 4 – Class I Railroad Fuel and Power Consumption for 1952

	Unit Fuel Consumption		
	Switching, per hr	Freight, Including Locomotive and Tender, per 1000 gross trailing ton-miles	Passenger Train, per car-mile
Steam-Coal, lb	899	116	20.6
Steam-Oil, gal	66.9	8.9	1.0
Diesel-Oil, gal	7.1	1.6	0.3
Electric, kw-hr	113	27.8	3.7
Approximate total cost of transportation fuel and power for 1952			$525,000,000
Approximate cost of fuel if all 1952 operations by steam-coal at $5.50			$820,000,000
Approximate cost of fuel if all 1952 operations by steam-oil at 4.5¢			$936,000,000
Approximate cost of fuel if all 1952 operations by diesel-oil at 9.8¢			$385,000,000
Approximate cost of power if all 1952 operations by electric at 10.52 mills			$642,000,000
Approximate cost of fuel if all 1952 operations by electric at 9.8¢			$ 16,000,000
Approximate total cost if all 1952 operations by electric			$658,000,000

with steam operation, when allowance is made for the operation and maintenance of the distribution system and the stationary converting plants of the 223,000 miles of mainline track.

It is interesting to note that the total production and importation of oil in the United States for 1952 was approximately two and one-half billion barrels of oil. If all railroad operation was performed with diesel power, the total quantity of distillate required would be only 3.75% of this total. If the complete railroad operation was with oil-fired steam power, the fuel requirements would be approximately 20% of the total fuel produced and imported.

The fuel requirements with oil-steam power are over 5.3 times that required with diesel power. This favorable ratio with diesel power is not explained by the relations between the optimum efficiencies of the two types of power. The large saving with diesel power is explained by the high efficiency of diesel locomotives over their total range of operation.

Another point of great importance in connection with liquid fuel supply and consumption is that a barrel of residual oil and distillate oil have practically the same Btu value. Where fuel must be imported or transported over a long distance and where stock piling is required, the quantity of fuel consumed may be more important than the actual unit cost of the fuel. In any case, the price of fuel, whether the fuel be residual or distillate, varies according to the demand and the balance in production to suit the most economical processing of the crude oil.

The question of fuel oil will be considered further when discussing the gas turbine locomotive.

Summary of Standard Locomotive Requirements

The factors that should control the design and required performance of standard locomotives in Class I railroads may be briefly summarized as follows:

(a) The road locomotive should consist of units that can be readily assembled to give horsepower outputs varying between 1500 hp and over 6000 hp.

(b) The axle loads and profile of the units should broadly comply with the permissible axle loads and clearances of practically all Class I railroads.

(c) The transmission capacity should be adequate to develop the locomotive horsepower rating fully over a large speed range to permit operation on level, rolling, and grade profiles.

(d) The locomotive should preferably have a good fuel efficiency over its normal operating range.

With these requirements in mind, we may proceed with a discussion of the gas turbine locomotives in revenue service today.

Popular Appeal of the Gas Turbine Engine

The possibility of obtaining power through the simple rotary gas turbine has certainly caught the popular imagination. The inherent simplicity of this engine is, of course, intriguing.

Its initial success employing waste heat to supercharge gas and diesel engines emphasized its economic possibilities. Its rapid development with turbojet and turboprop aviation engines has dramatized its extraordinary weight ratio with relatively large outputs.

The gas turbine today is as popular with mechanics as space travel is with small boys. A press release in 1951 by Sir Claude Gibb of the British Association for the Advancement of Science predicted that within 50 years all autos would be powered by gas turbines. All that would be required to bring this about is a *"compact, low-cost, efficient, regenerative heat exchanger,"* according to Sir Claude Gibb.

Fifty years is a reasonably long period, and in the interim we may rest assured of the continued development of the simple gas turbine for aviation and other applications to which it naturally applies. This development is largely the responsibility of those industries, the members of which largely compose this Society.

Stationary powerplants, marine, rail, and pipeline transportation and industries where waste heat is available are also interested in this development of the hour. However, unlike the supercharging and turbine-jet applications, which are obvious for the gas turbine, these other industries will have to give more thought to the conservation of our fuel supply and the general economic problems in the application of gas turbine power.

Development of Gas Turbine Locomotive

The development of the exhaust-gas-driven supercharging sets for aviation gasoline engines and diesel engines gave the necessary production and early metallurgical experience with fairly high temperatures required to develop the combustion gas turbine.

In 1936, the Sun Oil Co. in Philadelphia received from Brown Boveri and Allis-Chalmers the first of 30 gas turbines employed with the Houdry process. These gas turbines were fitted with oil combustion chambers for starting purposes.

This first commercial combustion gas turbine was further developed by the Brown Boveri Co. and an important and historic stationary unit was installed in the Swiss Neuchatel powerplant in 1939.

In 1941, Brown Boveri installed a similar but smaller unit in a locomotive that operated on the Swiss and French railroads during the war years. The first order for a gas turbine locomotive was apparently a redesign of the 1941 unit delivered by the Brown Boveri Co. to the Western Region in England in 1950.

There are many purely experimental gas turbine locomotives of various kinds proposed and some in actual test. Although these are, in many cases, interesting, they have not seen any actual revenue service that would justify their inclusion

Table 5 – Gas Turbine Locomotives in Service

1. Railroad	Swiss Federal Railways	Western Region	Western Region	Demonstrator	Demonstrator	Union Pacific
2. Turbine Builder	Brown Boveri	Brown Boveri	Metropolitan Vickers	Westinghouse	General Electric	General Electric
3. Locomotive Builder	Swiss Locomotive Works	Swiss Locomotive Works	Metropolitan Vickers	Westinghouse	General Electric	General Electric
4. Electrical Builder	Brown Boveri	Brown Boveri	Metropolitan Vickers	Westinghouse	General Electric	General Electric
5. Rated Hp of Locomotive	2200	2500	3000	4000	4500	4500
6. No. of Turbines	1	1	1	2	1	1
7. No. of Generators per Turbine	1	1	3	2	4	4
8. No. of Combustors per Turbine	1	1	6	12	8	6
9. No. of Stages per Turbine	6	7	5	8	2	2
10. Rpm of Turbine Unit	5200	5300	7000	8750	6900	6900
11. Type of Regenerator	Cross flow	Cross flow	—	—	—	—
12. Top Cycle Temperature, F	1110	1100	1300	1350	1290	1290
13. Efficiency at Full Load, %	16	15.8	18	16.9	16	16
14. Wheel Arrangement	1A-B-A1	A1A-A1A	C-C	B-B-B-B	B/B-B/B	B/B-B/B
15. Total Weight (Maximum), lb	203,000	266,890	290,000	494,000	530,000	550,000
16. Weight on Drivers, lb	142,000	177,920	290,000	494,000	530,000	550,000
17. Length Over All, ft-in.	53-10	63-0	66-9¼	77-10	83-8	83-6½
18. Wheel Base, Total, ft-in.	41-10	50-1	53-0	62-9	68-3	68-1
19. Wheel Base, Rigid, ft-in.	11-1	11-10	15-0	9-0	9-4	9-4
20. Width, ft-in.	9-6	9-3	9-0	10-2½	10-7	10-7
21. Height, ft-in.	13-11	13-4	12-10	15-0	15-4	15-5
22. Continuous Tractive Effort, lb	14,000	12,400	30,000	52,800	77,800	105,000
23. Speed at Continuous Tractive Effort, mph	47.5	64	30	23	18.2	12.9
24. Speed, mph	70	90	90	100	69.3	65
25. Railroad	Swiss Federal Railways	Western Region	Western Region	PRR-MKT-C&NW	PRR-Nickel Plate-UP	Union Pacific
26. Date Order Placed	March, 1939	October, 1946	July, 1947	Demonstrator	Demonstrator	December, 1950
27. Date First Operated	August, 1941	October, 1949	February, 1952	1949	Nov. 14, 1948	1952, 1953[a]
28. No. of Locomotives	One	One	One	One	One	Ten

[a] For further details, see Table 8, which is given later.

in our discussion of performance records.

The available and published performance records of gas turbine locomotives are limited to two Swiss, one English, and possibly seven delivered by the General Electric Co. to the Union Pacific Railroad. Data describing these units, together with the General Electric prototype of the Union Pacific units and an experimental Westinghouse unit, are shown in Table 5.

Gas Turbine Locomotives in Revenue Service

All the locomotives in Table 5 have the following general characteristics:

1. All locomotives use the open cycle.
2. All locomotives use axial-flow turbines and compressors.
3. All locomotive turbines follow designs in use in stationary plants.
4. All locomotives employ electric power transmission.

As shown by item 12 in Table 5, the top cycle temperature of the Swiss units is 1110 F, whereas the top temperature of the Metropolitan Vickers unit and American units is between 1350 F and 1290 F.

The weight and output characteristics of these six locomotives may easily be determined from Table 5.

The three European locomotives have a weight of 94 lb to 107 lb per hp. Their continuous tractive effort is between 7 and 10% of weight on drivers. Their ratio of maximum speed to continuous speed is 1.4 to 3.0. These are all very low values according to our standards, which we explained when discussing transmission ratings and requirements.

The Westinghouse demonstrator has a low weight ratio of 123 lb per hp and a continuous tractive effort of 10.5% weight on drivers, with a maximum to continuous speed ratio of 4. This unit is experimental and for passenger service.

The Union Pacific units were discussed with other types of power in Table 3, and will be referred to again later.

We will consider the application and performance records of the other five gas turbine locomotives separately.

Swiss Federal Railway Brown Boveri Unit

This 2200-hp unit built in 1941 is the first gas turbine locomotive placed in actual service. The single-shaft turbine with axial compressor and turbine has apparently established the most promising basis for all gas turbine locomotive developments to date. It employs a regenerator and one combustor to burn a selected type of residual fuel.

It is still operating without any major changes to its original design, although its operation was restricted during the war because of fuel supply limitations. The available operating records are meagre and presenting them here would serve no useful purpose, particularly since a unit similar in design delivered in England in 1950 will supply the operating information required.

English Western Region Brown Boveri Unit

This unit of 2500 hp, known as number 18,000, was completed at the end of 1949 and, as previously

Table 6 – Test of Western Region Gas Turbine Locomotive No. 18,000

Run No.	1	3	2	4
Date	11/9/51	13/9/51	12/9/51	14/9/51
Weight of Locomotive, tons	119.2	119.2	119.2	119.2
Load and No. of Vehicles, tons tare for No. of vehicles				
Paddington-Westbury	395 for 12	395 for 12	—	—
Westbury-Exeter	359 for 11	359 for 11	—	—
Exeter-Plymouth N. Rd.	239 for 7	239 for 7	—	—
Plymouth N. Rd.-Newton Abbot	—	—	228 for 7	227 for 7
Newton Abbot-Exeter	—	—	398 for 12	356 for 11
Exeter-Paddington	—	—	388 for 12	386 for 12
Distance – Actual, miles	225.2	225.2	225.2	153.2
Under Power, miles	182.2	189.4	185.6	120.2
Ton-Miles – Excluding Locomotive	78,000	78,000	82,400	53,600
Time – Booked Running, min	259.0	259.0	275.0	—
Actual, min	252.3	251.4	269.0	186.1
Overall – Including Stops, min	265.5	264.4	296.3	220.0
Actual Less Drifting and Idling – Under Power, min	209.9	204.0	213.4	175.6
Average Speed, mph	53.6	53.7	50.2	49.4
Work Done, db-hp-hr	2,520	2,824	2,621	2,135
Average Db-hp	721	831	736	729
Average Db Pull, tons	2.25	2.59	2.46	2.47
Oil – Total Used: Heavy, lb	4,960	5,230	5,450	4,340
Light, lb	224	223	250	186
Total, lb	5,184	5,453	5,700	4,526
Lb/Db-hp-hr	2.05	1.93	2.17	2.12
Oil – Total Used under Power: Heavy, lb	4,496	4,727	4,759	3,970
Light, lb	177	172	180	148
Total, lb	4,673	4,899	4,939	4,118

	Overall total	Under power	Overall total	Under power	Overall total	Under power	Overall total	Under power
Oil Rate – Heavy, lb/hr	1,121	1,282	1,210	1,350	1,102	1,337	1,181	1,353
Light, lb/hr	50.6	50.6	50.6	50.6	50.6	50.6	50.6	50.6
Total, lb/hr	1,176.6	1,332.6	1,260.6	1,400.6	1,152.6	1,387.6	1,231.6	1,403.6
Oil Rate, Drifting and Standby, Heavy and Light – Total	550.6		550.6		550.6		550.6	
Gross Calorific Value of Fuel Oil – Heavy, Btu/lb	18,530		18,580		18,680		18,740	
Light, Btu/lb	19,610		19,590		19,530		19,490	
Oil Consumption, lb/mile	23.02		23.20		24.2		28.3	
$\frac{\text{Work Done at Drawbar}}{\text{Total Heavy Fuel Oil Used}} \times \frac{100}{1}$, %	6.95		7.42		6.56		6.68	
$\frac{\text{Work Done at Drawbar}}{\text{Heavy Fuel Used under Power}} \times \frac{100}{1}$, %	7.67		8.20		7.51		7.29	
$\frac{\text{Work Done at Drawbar}}{\text{Total Heavy and Light Fuel Oil Used}} \times \frac{100}{1}$, %	6.66		7.09		6.25		6.39	
$\frac{\text{Work Done at Drawbar}}{\text{Total Heavy and Light Fuel Oil Used under Power}} \times \frac{100}{1}$, %	7.40		7.91		7.22		7.02	

stated, is a development of the 2200-hp Swiss unit built in 1941.

It was delivered in England early in 1950 and considerable information regarding its construction, performance, and initial troubles has been published in the technical press and lately in a paper by Dymond.[1]

As shown in Table 5, the gas turbine works at a maximum temperature of 1100 F with a pressure ratio of 3. The unit burns residual fuel with a viscosity of 950 sec, Redwood No. 1 at 100 F and a net calorific value of 17,400 Btu per lb. For starting purposes, gas oil is employed. A Saurer diesel engine of 150 hp drives a 40-kw generator for lighting and auxiliary supplies. A train heating boiler of 1000 lb per hr rating for car heating is employed. This unit was given the usual shakedown operation before being placed in revenue passenger service in the middle of May, 1950.

During its first year of operation, the initial difficulties were those that might be expected with any new unit. After about 18 months, information was acquired regarding maintenance technique, with particular reference to the necessity of cleaning blading and intercooler tubes at frequent intervals.

Traction motor trouble and a heat exchanger fire reduced the service availability of the unit during its first two years of operation.

From its arrival in England early in January, 1950, to early February, 1952, or approximately for a period of two years, the unit completed 72,590 miles with a turbine operation of 1913 hr.

Due to certain blading defects, the unit was withdrawn from service in September, 1952, and the opportunity was then taken to overhaul the unit and install a new boiler. Up to Sept. 6, 1952, or for a period of about two and one-half years, the unit completed 117,789 miles and the turbine operated 3296 hr with an availability factor of 63.9%.

This unit was confined in revenue service to the haulage of high-speed passenger trains with relatively few stops. Table 6, which is from *Diesel Railway Traction*,[2] gives a test record of unit 18,000. The same article brings out the fact that,

[1] "Operating Experiences with Two Gas Turbine Locomotives," by A. W. J. Dymond. Presented before Institution of Locomotive Engineers, London, Feb. 18, 1953. Reviewed in *Oil Engine and Gas Turbine*, Vol. 20, March, 1953, pp. 401-402: "Service History of Western Region's G.T. Locos."

although the thermal efficiency at full load and 1100 F was 16.9% with a fuel consumption of 0.87 lb per hp-hr, at half load the temperature was only 650 F and the average thermal efficiency at the drawbar on the test runs never exceeded 8%.

As an example of the effect of load on the efficiency of the gas turbine locomotive, the same article points out that, with trailing loads of 200 tons, 300 tons, and 400 tons, which could be hauled on level track at 80 mph, the respective fuel consumptions were 0.09 lb, 0.075 lb, and 0.065 lb per ton-mile, corresponding to 1.95 lb, 1.6 lb, and 1.4 lb per drawbar horsepower. These values indicate that, with this particular unit, with a reduction in load of 50%, the unit fuel consumption per ton-mile is increased by 40%.

Dymond,[1] in commenting on the fuel consumption rates in these tests, remarked as follows:

"In this the corresponding figure of oil fuel consumed varies from 23.02 lb per mile to 28.3 lb per mile. With the specific gravity of the oil at 0.94 (average) at normal temperatures, this is from 2.47 to 3.00 gal per mile. As these values are substantially of the same order as the annual figures, it is reasonable to assume that the ratio of oil per db-hp-hr (1.93 to 2.17 lb) obtained on the tests obtains throughout the operating period. This is the equivalent of an overall work to fuel ratio of about 6.7%. Compared with the optimum efficiency at full load of about 16.8% obtained on the bench tests of the power unit at Baden, this is perhaps surprising. The explanation lies in the average power required of a locomotive being in the vicinity of 50% or less of full power, combined with the strongly drooping characteristic of the efficiency curve for the gas turbine. In this connection, idling and light engine running account for far more than their due share of fuel consumption."

Finally, on the basis of equivalent heat value of coal and oil, Dymond indicates that the equivalent coal fuel consumption of the gas turbine locomotive is between 65 and 70% of the run-of-mine steam-coal consumption.

The lubricating oil consumption of this unit is most favorable. Maintenance data due to numerous alterations and adjustments are of course, of no particular interest for this unit up to this date.

English Metropolitan Vickers Unit

Details of this 3000-hp unit known as 18,100, are shown in column 3, Table 5. It will be noted that the electric transmission employs motors on all six locomotive axles, and, even with this liberal transmission according to European practice, it has the unusual weight ratio of 96.5 lb per hp.

As indicated in Table 5, it works at a maximum temperature of 1300 F with an optimum efficiency of 18%. It burns distillate similar to that employed with diesel engines.

This unit was delivered at the end of 1951 and went into shakedown service early in January, 1952. These runs included the hauling of unusually heavy trains.

The first revenue trip was made early in April, and it was in regular revenue operation up to the middle of September, 1952, when the unit was taken out for revisions to control circuits and other minor troubles.

In this operation of approximately eight months, the unit ran 42,000 miles and the turbine operated 1237 hr. The fuel consumption obtained with this unit averaged approximately 3 gpm and, according to Dymond, the efficiency in comparison with steam locomotives was similar to that already outlined for the Brown Boveri unit. Up to August, 1953, 116,429 miles were operated, during which time it was in service approximately 10 months, with an overall load factor of 58.2%.

The period of operation was altogether too short to indicate any trend in maintenance as various modifications are still being carried out with the view to limiting maintenance to a minimum.

English Locomotives 18,000 and 18,100

In the general discussion of Dymond's paper,[1] certain interesting points were emphasized. The two gas turbine locomotives apparently outperformed the steam power that they would normally replace.

The chief difficulty with the Brown Boveri unit 18,000 is the short life of the combustion-chamber lining. The distortion and wasting of this liner, according to Dymond, has pointed consistently to the difficulty of obtaining satisfactory combustion of residual fuel over the entire range of operation. Arrangements have been made with the Shell Petroleum Co., in co-operation with the locomotive manufacturer and the railway executive, to develop combustion chambers and a method of combustion to overcome these difficulties.

Dymond also pointed out that, in connection with the traction motors of unit 18,000, the motor design emphasized speed rather than tractive effort.

In commenting on Table 5, we have shown that the flexibility of the motor transmission system with this unit is very limited, giving a ratio of only 1.47 between maximum and continuous speeds, whereas with diesel locomotives in freight service in the United States we employ ratios up to 7.

Other speakers, commenting on Dymond's paper, point to the short life of the flame tubes of the distillate burning unit 18,100, but indicate that improvements have been obtained with new materials.

It was also pointed out that in railroad application a good thermal efficiency would be required at 20 to 30% of the maximum output and that the efficiency at full load was of little significance.

P. Draper of Shell amplified these comments by

[2] See *Diesel Railway Traction*, Vol. 6, July, 1952, pp. 179-181: "Tests of British Gas Turbine Locomotive."

stating that, even if the locomotives were trouble-free, they could never make a really good case for gas turbines on railways because of their inherent design limitations.

GE and UP Gas Turbine Locomotive Ratings

The ratings for the two gas turbine locomotives employed in England are presumably at sea level and an ambient temperature between 60 and 70 F. In the case of the gas turbine locomotives delivered to the Union Pacific, it is necessary to consider the rating of these units in relation to temperature and elevation, due to the large range in both temperature and elevation in which these units must operate.

Table 7 gives the rating for the units over the range of temperature that may be expected and the range of elevation that may be encountered. In Table 7 it will be noted that there is a variation in the gas turbine output of 300 hp for each 10 deg change in temperature. There is a variation in the output of the gas turbine engine of 100 hp for each 1000-ft change in altitude.

The normal rating of the gas turbine unit is 4750-hp gross output with 4500-hp input to the generator at 80 F and 1000-ft elevation.

At 6000-ft elevation, which is the approximate average elevation on the Green River to Ogden operation, and at 80 F, the turbine power output is 4250 hp with 4000-hp input to the generator.

For the purposes of a practical electric transmission system, it is understood that the generator is designed for a maximum output of approximately 6000 hp.

Due to the possible variation in horsepower caused by temperature or elevation changes, the characteristic curve of this gas turbine electric locomotive must be represented by an envelope covering the extreme ranges of possible operation.

Fig. 1 gives the characteristic operating range for the gas turbine locomotive, the cross-hatched area showing the range of operation between the maximum output of approximately 6000 hp at 20 F and 6000-ft elevation and the output of 4150 hp at 100 F and 1000-ft elevation. This hatched area might be extended to include a 3550-hp output at 120 F and 1000-ft elevation.

For the purpose of comparison with other types of railroad motive power, Fig. 1 also includes curves showing the output for a 3-unit 4800-hp diesel locomotive, a 4-unit 6400-hp diesel locomotive, and the 6000-hp ignitron locomotive. The weight and adhesion characteristics of all these locomotives are shown in Table 3 and were discussed when considering locomotive electric transmission.

For a complete understanding of the locomotive curve shown on Fig. 1, we refer you again to Table 3. The gas turbine locomotive with eight driving axles and traction motors has a continuous tractive effort of 105,000 lb.

From Fig. 1 it will be noted that this tractive effort may be delivered at a speed varying from 11.6 to 16 mph, depending upon atmospheric conditions.

The 4800-hp diesel locomotive, with 12 traction motors and driving axles, has a continuous tractive effort of 157,500 lb at a speed of 9.3 mph, which,

Table 7 – Rating Table – Gas Turbine Locomotives for Union Pacific Railroad

Traction Hp	Turbine Hp	Air Temperature, F	Altitude, ft
3300	3550	120	1000
3400	3650	100	6000
3900	4150	100	1000
4000	4250	80	6000
4500	4750	80	1000 (standard rating)

See Fig. 1 for locomotive output, as the generators cannot utilize the full output of the turbine, as shown in the following ratings:

4600	4850	60	6000
5200	5450	40	6000
5800	6050	20	6000
6400	6650	0	6000

Note: Above based on:
300-hp change for 10 deg air change.
100-hp change for 1000-ft altitude change.

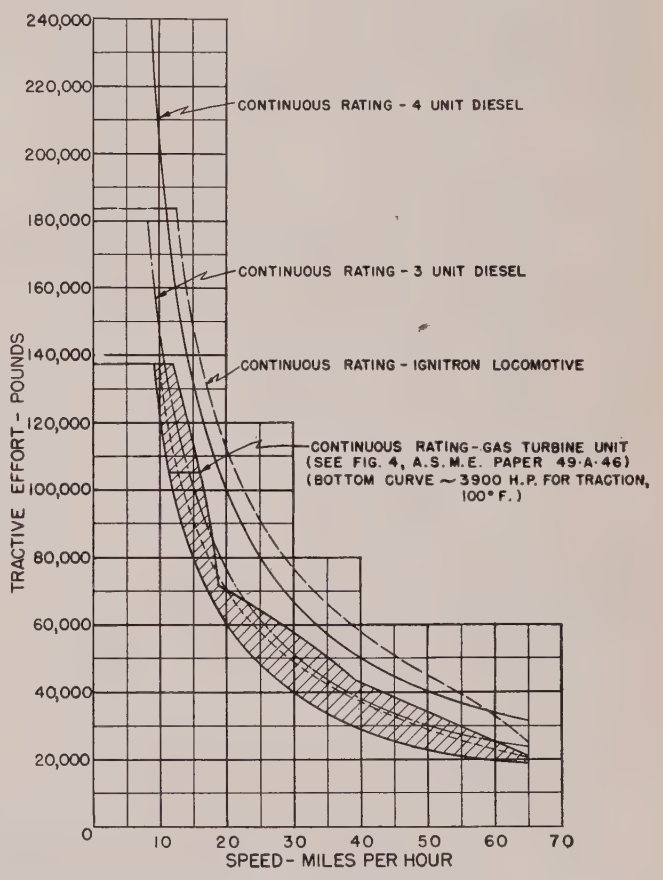

Fig. 1 – Tractive effort speeds curve of electric, diesel, and Union Pacific gas turbine locomotives

as already mentioned, is 50% greater than that of the gas turbine locomotive.

The 6000-hp ignitron electric locomotive, with 12 traction motors and driving axles, has a continuous tractive effort of 132,000 lb. The corresponding 6400-hp diesel locomotive has 16 traction motors and a continuous tractive effort at 18% adhesion of 180,000 lb, which is 40% greater than that of the electric locomotive.

We have already explained, when discussing the electric transmission, the relation between the maximum and minimum permissible speeds and the corresponding flexibility of the various units. These values are given in Table 3.

Fuel Consumption with UP Gas Turbine Locomotives

The next point of interest, in connection with the gas turbine locomotive, is its fuel consumption with various horsepower outputs. The 4500-hp Union Pacific locomotive full-load fuel consumption is stated to be between 550 and 600 gal per hr of 18,600 Btu per lb fuel. At idling, the fuel consumption is stated to be approximately 200 gal per hr. These values are assumed in Fig. 2, which gives the fuel consumption of the locomotive unit with its normal traction output at 80 F and 6000-ft elevation.

For comparative purposes, the corresponding fuel consumption curve for a typical 3-unit diesel locomotive, with a maximum output of 4800 hp, is also shown in Fig. 2.

With the rated output at 80 F and 6000-ft elevation of 4000 hp for traction, the gas turbine fuel consumption is 550 gal per hr and the diesel fuel consumption at 4000 hp is 250 gal per hr. The ratio of fuel consumption at this load is 2.2 to 1.0. While idling, the fuel ratio is 8.0 to 1.0.

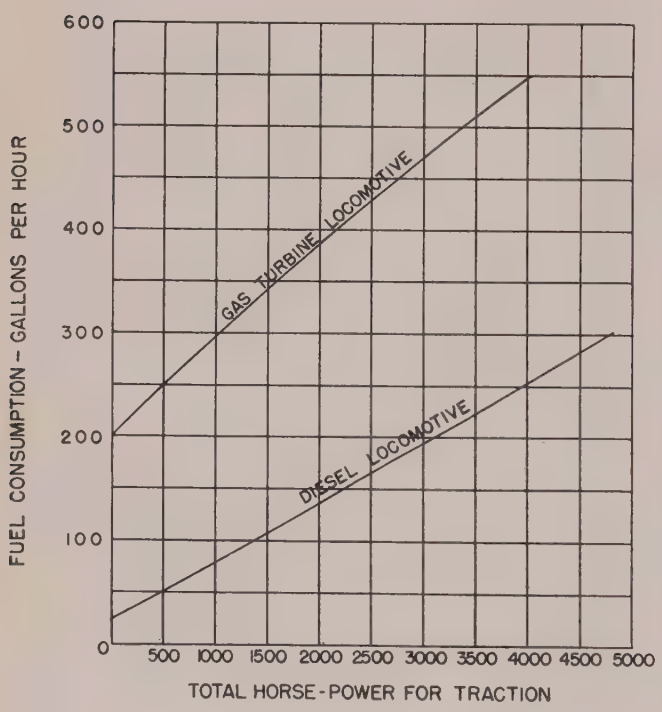

Fig. 2 – Approximate fuel consumption for Union Pacific gas turbine locomotive at Green River (6000-ft elevation, 80 F air temperature) and standard 3-unit diesel locomotive

The average fuel consumption rate is given as 335 gal per hr for the gas turbine locomotive. The corresponding diesel rate would be approximately 100 gal per hr, which gives a fuel ration of 3.35 to 1.0.

These values are significant and point to the fact that the gas turbine locomotive on the Union Pacific would, under normal railroad conditions of operation, consume at least three times the fuel required with a diesel locomotive in comparable service.

With commendable foresight, the Union Pacific has arranged its locomotives so that dead-head movements from engine house to train and certain switching movements can be accomplished with a small diesel motor-generator set through a few traction motors. The locomotive is also arranged so that the gas turbine can be closed down and started up from the control position of the operator. With this arrangement the turbine can be closed down on long coasting runs, with a great saving in fuel consumption. With the turbine closed down, the diesel motor-generator supplies the compressors and other required auxiliaries.

The extent to which the small diesel motor-generator set is employed is explained by the 1000-gal diesel fuel tank carried on this locomotive. This is nearly 30% of the normal fuel capacity of a 4800-hp diesel locomotive.

In actual operation in 1952 on the Green River to Ogden runs, the published fuel consumption for units 51 and 52 is given as 14 to 15.3 gal of residual oil and 0.55 to 0.88 gal of diesel oil per locomotive-mile. The fuel consumption is also given as averaging 4.28 gal per 1000 gross trailing ton-miles.

On this run of 175 miles, the first 65 miles east of Ogden is a helper section of approximately 1.14% grade. This relatively long stretch of helper service naturally raises difficulties in assigning fuel on a ton-mile basis, and information is not readily available concerning diesel fuel consumption on this division.

However, the 1952 national average of diesel fuel consumption and steam locomotive bunker C consumption, including locomotive and tender, was 1.6 gal and 8.9 gal respectively. Allowing for the locomotive weights, the national fuel consumption on a trailing basis would be approximately 1.75 gal per 1000 gross trailing ton-miles for diesel power and 10.2 gal with steam power. At these rates, the gas turbine fuel ratio with diesel becomes 2.45 to 1.0 instead of the 3.35 to 1.0 ratio indicated in Fig. 2. As against the steam-oil national average, the ration is 0.43 to 1.0. When English engineers compared steam-coal power with their two gas turbine locomotives, they found that the relation was from 0.6 to 0.7 to 1.0.

We may therefore conclude that, with carefully

supervised operation, where advantage is taken of assigning maximum economic tonnage to the gas turbine locomotive, with a minimum of idle turbine running, the fuel consumption has been reduced approximately in the ration of 3.35 to 2.45 or by more than 25%.

It is quite possible that the fuel consumption with the gas turbine on a comparatively level or rolling profile, on a ton-mile basis, will be found to be greater than on the heavy grade section where full load, with minimum turbine idling time, can be obtained.

The auxiliary boilers for passenger-car train heating employed with electric, diesel, and gas turbine locomotives generally use distillate or diesel fuel. Although this is not a requirement on the Union Pacific freight locomotives, it nevertheless represents over 30% of the fuel consumption with diesel passenger locomotives during the winter months and averages throughout the year between 16 and 17%.

UP Gas Turbine Fuel Specification

At the higher temperatures of operation required with gas turbines, to obtain good thermal efficiency (between 1200 and 1400 F), it would be advantageous to burn distillate or diesel fuel. It becomes immediately apparent, however, that, with the fuel consumptions experienced on the Union Pacific and in Europe, the cheapest fuel possible is essential, with due consideration for maintenance and reliability.

The Union Pacific Railroad is, I understand, in the fortunate position of having readily available a supply of residual bunker C fuel at relatively low rates, although the diesel fuel apparently costs more than in other districts.

Early experience in this country and in Europe, with all types of gas turbine powerplants, clearly indicated the necessity of carefully controlling the vanadium, calcium, and sodium salts normally present in most residual oils. This problem is being solved on the Union Pacific by the cooperative action of the builder, oil supply company, and the railroad – and all residual oil is supplied to a definite specification.

This specification has already been discussed by Buckland.[3]

Operating Record of UP Gas Turbine Locomotives

Table 8 gives the record of the deliveries of the first 10 locomotives. For design data, reference may be made to Table 5.

The first six units have, according to the latest record available, operated an average of approximately 9000 miles per month. The cumulative mileage to January, 1953, was 437,771.

Table 8 – General Electric – Union Pacific Gas Turbine Locomotives

Locomotive No.	51	52	53	54	55
Date delivered	Jan. 31, 1952	Apr. 9, 1952	May 7, 1952	June 4, 1952	July 9, 1952
Where	Green River	Green River	Green River	Green River	Green River
Locomotive No.	**56**	**57**	**58**	**59**	**60**
Date delivered	Aug. 13, 1952	May 20, 1953	July 3, 1953	Aug. 12, 1953	Aug. 27, 1953
Where	Green River	Los Angeles	Green River	Green River	Green River

Locomotive No. 51, from January, 1952, to June, 1953, operated 157,315 miles. Locomotive No. 52, from April, 1952, to June, 1953, operated 142,010 miles. Availability during the above period was 80% for No. 51 and 82.4% with No. 52.

These statistics indicate a very fine performance for new units, which is confirmed by an order for a further 15 locomotives of similar design.

The 10 gas turbine locomotives are pooled with 40 diesel units, which are assembled in 2-, 3-, and 4-unit locomotives, as well as with a number of oil-burning steam locomotives. The gas turbine locomotives take their maximum tonnage as far as possible and the diesel locomotives vary in size, according to the tonnage available.

The gas turbine locomotives are maintained with practically the same maintenance facilities as have been installed for diesel locomotives.

It is claimed one spare turbine unit is available to protect the 10 locomotives and an additional set will be available when the next 15 are delivered.

Combustion chambers are inspected at 100-hr intervals and nozzles are inspected and some replaced at each 50-hr inspection. Control simplification is in process and the maintenance of the electric equipment is reported to be the same as with diesel locomotives.

No data on the out-of-pocket operating costs of the 4500-hp gas turbine have been published. One would naturally wish to compare such data with data for a typical 4500-hp diesel locomotive. The indications are that the crew wages with the diesel locomotive might be less if its extra hauling capacity was exploited. With existing fuel costs there is, with the Union Pacific method of operating, probably only a small advantage with diesel power. In a final analysis, possibly the only difference in operating cost may be the maintenance cost of the two types of power, and this difference can only be between the gas turbine engine and the diesel engine, as the transmission equipment is similar in both cases.

It would appear that some time must elapse be-

[3] See ASME Paper No. 52-A-16, presented at Nov. 30-Dec. 5, 1952 meeting: "Residual Fuel Oil Ash Corrosion," by B. O. Buckland, C. M. Gardiner, and D. G. Sanders.

Table 9 – Westinghouse Gas Turbine Locomotive Performance Data

Railroads	PRR	M-K-T	C&NW
No. of Trips Scheduled	59	103	138
No. of Trips Made	55	101	120
Availability for Assigned Runs, %	93	98	87
Road Delays	None	1	5
Average Length of Train – Cars	26-29	11-15	12-17
Train Miles	7100	27,775	25,380
Gal Oil per Car-Mile (approximate)	0.23	0.53	0.57

fore a definite answer is obtained on this question.

Gas Turbine Locomotive Possibilities

Although the first commercial gas turbine locomotive was placed in service about the same time that the first commercial diesel freight locomotive entered service in 1941, its development has been slow, due principally to the greater metallurgical problems with the gas turbine.

The relatively large Union Pacific application is the most important. It will be followed with great interest.

It is evident that the economic possibilities of the gas turbine as developed to date in railroad service will depend on the employment of cheap fuel and will, in consequence, be confined to locations where fuel shipping cost is a minimum.

There are, of course, many promising developments with gas turbines. The powdered-coal gas turbine with coal at $5.50 per ton would have a fuel cost less than a diesel locomotive with distillate at 10¢ per gal. The only available performance data of 1952 for the Westinghouse gas turbine locomotive is shown in Table 9.

The free-piston or Pescara engine is claimed to have a higher thermal efficiency than the simple type employed in railroad service today. In countries limited to peat fuel, the employment of this type of engine in a gas turbine may give very good economic results.

Finally, the normal production of jet and turbojet engines must result in metallurgical developments that will favorably affect all gas turbines.

The application of the gas turbine locomotive to our Class I railroads will depend entirely on its economics. If the gas turbine locomotive should prove cheaper to operate than other types of power, it will prevail.

Limited Field for Special Locomotives

We have shown however, that the number of locomotives required to operate our railroads is quite limited and the horsepower required is surprisingly small. It is, therefore, essential that, in the introduction of a new type of power, it should be arranged for general or universal use and not limited to special applications.

It is fortunate that the gas turbine locomotive can employ the same traction motor transmission developed as standard for diesel locomotives. If, however, the gas turbine locomotive and straight electric locomotive, of comparable outputs, employ some 50% less transmission capacity than the established standard diesel locomotives, they are simply developing units with special and limited applications.

In order to cover the field of general application on our Class I railroads, a considerably smaller unit than the 4500-hp Union Pacific locomotive will be required. It certainly would not be economic to employ two 4500-hp Union Pacific gas turbine locomotives to obtain the hauling capacity of a 4-unit 6000-hp diesel locomotive.

The significance of developing units that can be universally employed is primarily that, in order to meet competition with diesel and straight electric locomotives, a broad market with mass production must be created. Special applications, with custom-built units, will not create this condition, but will tend to revert to the old steam and electrification practice, where every application was a special study in itself with no possibility of power interchange, even on connecting railroads.

Acknowledgments

In concluding, I must acknowledge with thanks receipt of information included in this paper from the following sources: F. Fahland, general mechanical engineer, Union Pacific Railroad, who has supplied a large part of the operating data discussed; R. A. Williamson, A. H. Morey, and J. E. Wilson, General Electric Co., for published details and developments of the gas turbine locomotive; our English and Swiss friends for the published information of their locomotive design and operation.

I also made extensive use of *Diesel Railway Traction,* a London publication, which contains much information on the design and operation of European units.

Additional References

1. ASME Paper No. 49-A-46 presented at Nov. 27-Dec. 2, 1949 meeting: "4500-hp Gas Turbine Electric Locomotive," by A. H. Morey and R. A. Williamson.

2. ASME Paper No. 52-A-56, presented at Nov. 30-Dec. 5, 1952 meeting: "Gas Turbine Electric Locomotives on Union Pacific Railroad," by A. H. Morey and F. Fahland.

3. ASME Paper No. 52-A-127, presented at Nov.

30-Dec. 5, 1952 meeting: "Use of Residual Fuel Oils in Gas Turbines," by P. Draper.

4. AIEE Technical Paper No. 53-69, presented at AIEE Winter General Meeting, Jan. 19-23, 1953: "Electric Transmission with Diesel Locomotives," by P. A. McGee.

5. ASME Transactions, Vol. 75, February, 1953, pp. 121-234: "Gas Turbine Progress Report – 1952."

6. Association of American Railroads Circular No. D.V. 1298 (1953), "Mechanical Division Report on Locomotive Construction."

7. *Mechanical Engineering,* Vol. 65, April, 1943, pp. 261-266: "Gas Turbine Locomotive with Electrical Transmission," by P. R. Sidler.

Additional Note

Since the presentation of my paper and the above discussion – on Nov. 3, 1953 – the Union Pacific Railroad has announced the placing of an order for 205 diesel locomotives, which were reported to cost them $35,769,410. The announcement emphasizes the confidence of this railroad in the future and the complete dieselization of the mainline of this road between Omaha and the Pacific Coast.

Table A – 2200-Hp Swiss Gas Turbine Locomotive in Passenger Service

Fuel Consumption for 220-Ton Train (Locomotive Not Included), gal per 1000 ton-miles	7.07
Fuel Consumption for 110-Ton Train (Locomotive Not Included), (Frequent Stops), gal per 1000 ton-miles	17.7

Note: In both cases, fuel weighed 7.2 lb per gal.

Table B – Operating Results with English Brown Boveri Unit No. 18,000 in Passenger Service

	Run No. 1	Run No. 3	Run No. 2	Run No. 4
English Ton-Miles	78,000	78,000	82,400	53,600
U. S. A. Ton-Miles	86,000	86,000	98,050	59,000
Total Fuel, lb	5,184	5,453	5,700	4,526
Lb per 1000 Ton-Miles	60	63.4	63	76.6
Gal per 1000 Ton-Miles	8.1	8.6	8.5	10.35

Note: Average weight of fuel was 7.4 lb per gal.
(For additional data see Table 6 of main paper)

Table C – U. S. A. 1952 Values Listed for Comparison Purposes

Passenger Service – Steam-Oil, gal per 1000 gross ton-miles	13.35
Passenger Service – Diesel, gal per 1000 gross ton-miles	4.00
Freight Service – Steam-Oil, gal per 1000 gross ton-miles	10.20
Freight Service – Diesel, gal per 1000 gross ton miles	1.75
Freight Service – UP – Gas Turbine, gal per 1000 gross ton-miles	4.28

DISCUSSION

Additional Data on Swiss Gas Turbine Units

– Adolph Meyer
Brown Boveri Corp.

TABLE A gives operating results with the original 2200-hp Swiss gas turbine locomotive, the first gas turbine locomotive to be placed in revenue service. This locomotive has operated approximately 220,000 miles in 100 months. Its average train weight for this period has been 330 tons, with a gross ton mileage of 66,000,000.

This locomotive has operated on the following lines: Basle-Chur, and as part of this: Basle-Zurich, Basle-Luzern, Basle-Delsberg. For four months the locomotives ran on the French National Railway, Basle-Chaumont, and for six months on the German State Railway, Munich-Würzburg.

For train heating in the wintertime, the Swiss locomotive produces about 220 kw, which is actually excess power obtained because of cold-weather conditions. Thus, fuel consumption does not increase, as compared to that of summer operation. Winter mean temperature is about 15 F and summer mean temperature is about 70 F.

The Swiss Federal Railways takes the locomotive out of service every two months for two days for the following:

1. Cleaning of the air heater.
2. Cleaning and eventually replacing the fuel nozzle (only one nozzle up to the biggest sizes of gas turbines and Velox boilers).
3. Inspection of the combustion chamber.

In gas turbine stations these items cost about 0.035 centimes per kw-hr, or 0.01¢ per kw-hr.

There is a general overhaul by the Swiss Federal Railways every 125,000 miles. Parts to be replaced include:

1. Fuel nozzle (erosion).
2. Inner (guide) parts of the combustion chamber. Of 250 parts, 40 had to be replaced in 14,000 hr of running.
3. Once the air heater was burned out after service during a night in the depot. Since then the cleaning is done every two months.

It might be noted that the highest maintenance costs occur for the replacement of the inner parts of the combustion chamber.

Table B gives operating results with the English Brown Boveri Unit No. 18,000 in passenger service.

Table C gives 1952 values for American units, for comparison purposes.

I might add that the largest gas turbine locomotive we would be in a position to build with one turbine and one combustion chamber, together with the heat exchanger, would be approximately 5000 hp. We would, however, prefer to build the 2500-hp size now operating on the British railway.

Gas Turbine Locomotive Has Many Attractive Features

– Walter C. Hadley
Automotive Engineer

THERE is no possibility that the gas turbine locomotive could replace the diesel in the way that the diesel has recently replaced steam. For one thing, practically all

Class I railroads are now well equipped with nice new diesel locomotives and new shops to care for them, all representing a very large investment. For another, the possible savings in going from diesels to gas turbines are nothing like as large as the savings in going from steam to diesel. None the less, the gas turbine has many attractive features, which should find it a useful place in many railroad applications, once its teething troubles have been brought under control. Two of these are:

1. Ability to use cheap fuel. While present gas turbine fuel requirements are far from clear, there can be no question that it will be a great deal easier to burn cheap fuels in the continuous turbine combustion process than in the intermittent diesel combustion process, where a cylinderful of fuel and air has to burn completely in less than 1/25 sec.

2. Decreased maintenance. While gas turbines are not getting any simpler, it is not likely that they will ever approach the very large number of precision parts found in the diesel, most of which reciprocate and are subject to all kinds of wear and breakage. This can only mean less maintenance in favor of the turbine.

After these general comments, the following remarks are offered with respect to specific items in the paper:

Fuel Consumption

From the data given in Table 4, it can be calculated that, if the Class 1 railroads had all been dieselized, their 1952 requirements would have been approximately 93,000,000 bbl of diesel fuel. While it is quite true that this would be less than 4% of the total crude being processed per year for U. S. domestic use, it would, nevertheless, account for about one-fifth of all the distillate fuel produced from that crude, and very likely for about two-fifths of all the diesel fuel produced. On the average, about 20% of the crude run appears in the form of distillate fuel, about half of which could be expected to meet railroad diesel fuel specifications. This distillate fuel can be, and is, used for home heating, automotive diesels, and other larger diesels, such as railroad, marine and power plant engines. It can also be processed in refineries to produce additional gasoline. In addition, during the past few years, a part of it is finding its way into the fuel used in jet aircraft. It is, accordingly, a highly sought after portion of the crude oil barrel.

It can also be calculated from Table 4 that, if all the Class 1 railroads had used oil-fired steam locomotives during 1952, they would have required about 495,000,000 bbl of residual fuel. As stated in the paper, this would account for about 20% of the total U. S. crude run. It would also account for virtually all the residual fuel produced, since, on the average, about 20% of the crude run appears in the form of residual fuel. (In actual fact, the railroad use of residual fuel would not have reached this figure because, if all the railroads used steam locomotives, at least half of them could logically be expected to burn coal.) Compared to distillate fuel, residual fuel finds comparatively few uses. It can be, and is, burned under boilers in power plants, ships, and steam locomotives. It can be used, with limitations, in large diesel engines if the operator will go to sufficient trouble in the matter. It can be processed, in a few refineries, to make gasoline and coke or to make asphalt. Compared to distillate fuel, it is not eagerly sought after, and is frequently disposed of at a lower price than was paid for the original crude.

From the standpoint of conservation of oil resources, it would appear more desirable for the railroads to burn the relatively useless residual fuels, either in gas turbines or steam boilers, than to burn a smaller quantity of the highly useful distillate fuels. It would, of course, be better yet for them to burn nothing but coal.

From the standpoint of economy, it would obviously be desirable to concentrate on the use of residual fuels, either in gas turbines or diesel engines, since by the laws of supply and demand this product is bound to remain much cheaper than distillate fuels. As mentioned earlier, it seems likely that the gas turbine will be ready to accept this diet before the railroad diesel engine will.

In the discussion of the Union Pacific gas turbine locomotive, it was pointed out that the unit was quite susceptible to the effects of inlet air temperature and pressure. It is, of course, true that the power developed by a gas turbine falls off quite rapidly as the air inlet temperature is increased, and that the turbine is more sensitive than the diesel engine in this respect. On the other hand, the power increases with equal rapidity when the air inlet temperature is lowered, and it is unfortunate that the transmission arrangements were not designed to take advantage of the extra power available in cold weather.

In order to compensate for power loss with air temperature rise, it might be worth while to consider a suggestion presented by Mr. Gatewood of the Elliott Co., in connection with marine gas turbines, which was that a waste-heat boiler be used to produce low-pressure steam to be bled into the turbine inlet, providing a considerable gain in power and also some improvement in thermal efficiency.

With respect to the effect of air inlet pressure, as governed by altitude, it seems likely that the diesel and the turbine would be equally susceptible, and that the 6400-hp diesel shown in Fig. 1 would be developing closer to 5400 hp at 6000-ft elevation.

Finally, it is basic that both the power and the efficiency of the gas turbine are functions of the turbine inlet temperature, and it is quite evident that this temperature is climbing slowly as more is learned about metallurgy and turbine design. The turbines listed in Table 5 show a rise from 1100 F to 1350 F, while published data on the most recent 4500-hp Metropolitan-Vickers design for use in small British naval vessels show a maximum inlet temperature of 1480 F. This turbine, of course, is not designed for a very long life; however, as further progress is made it is likely that turbine fuel consumptions will come closer and closer to those of railroad diesels.

Favors Large Single-Unit Gas Turbine Locomotive

— F. Fahland
Union Pacific Railroad Co.

THE comparison of horsepower requirements per ton of load handled by various means, showing that the railroads use only 1-1½ hp per ton for high-speed freight, is proof that the railroads are a very economical means of transporting freight.

The statements throughout the paper that the 1500-hp diesel unit usable in multiples is best suited for American railroads and that these units are superior to a single-unit gas turbine electric locomotive would appear to be made for consumption of the diesel locomotive builders. The size of the 1500-hp diesel unit was apparently established by the fact that the diesel-engine builders have been unable to produce a larger high-speed diesel suitable for railroad service. This small engine has forced the railroads to use multiple units, with their multiple maintenance problems. Maintaining three diesel locomotive units compared to one single-unit locomotive, in our opinion, is like maintaining three automobiles in place of one. If anyone thinks maintaining three automobiles is as economical as maintaining one, he should try to prove it to himself.

Recently, we had four diesel units jackknife, breaking 234 ft of rail. This is trouble we would not expect to have

with a single-unit locomotive such as the gas turbine electric locomotive.

The handling, turning, and directional testing of multiple units at terminals involve work, time, and costs in excess of that required to handle a single unit. To turn three units they must be run over a wye or disconnected, handled as one or two units, and reconnected.

The author places considerable emphasis on the fact that three diesel units of 1500 hp each, totaling 4500 hp, with continuous tractive effort of 157,500 lbs should haul 50% more tonnage on a given grade than the 4500-hp gas turbine electric locomotive with continuous tractive effort of 105,000 lb. The diesel locomotive, with its low horsepower per weight and its ability to drag just about all the tons it can start, has a shortage of horsepower for acceleration of trains and has become known on the railroads as a drug locomotive. The gas turbine electric locomotive, with its higher horsepower per weight, can accelerate any train that it can start.

The railroads today are conscious of the necessity for expediently handling freight, and we have found that the lighter gas turbine electric locomotives accumulate as many gross ton-miles of freight or loads hauled as the diesels. In other words, they do equally as much work as three diesel 1500-hp units.

The author makes the statement that it is his understanding the Union Pacific is in the fortunate position of having readily available a supply of residual bunker C fuel at relatively low rates. I wish to elucidate this statement. Fuels used in the gas turbine electric locomotives have been purchased at the current market price for the territory in which the fuel is obtained. We have found no way of obtaining fuel oils from the oil companies at less than market price. It has been necessary for us to purchase bunker C modified fuel on the West Coast where cost of such fuel is the highest on our railroad. Even with the use of this high-priced fuel, the cost of fuel for the gas turbines compares favorably with fuel cost for the diesels on a 1000 gross ton-mile basis. The author has compared the quantity and cost of fuel used in the diesel locomotives with that used by the steam and gas turbine locomotives. Fuel costs alone are only a part of the cost of operating a locomotive. Cost comparisons should include lubrication and maintenance costs. Some of the steam locomotives were maintained in operating condition for 50 years. I wonder what the maintenance cost would be on a 50-year-old diesel-electric locomotive, or how frequently must the diesel and electrical parts be replaced. We in the mechanical department of railroads believe that the cost of maintaining all locomotives, including the diesels, is too high and it is our responsibility to try and find ways of reducing these costs.

I do not wish to convey the impression that the diesel locomotives have failed to do a good job for the railroads. However, there is still room for improvement in railroad motive power, and it is possible that the single-unit gas turbine electric locomotive would be the most satisfactory and economical locomotive for mainline high-speed freight service.

Questions for The Author

— Edson Barlow

Transportation Research & Development Station

I WOULD like to ask the author a few questions, as follows:

1. Mention is made of arrangements made with the Shell Petroleum Co. to develop combustion chambers and methods of combustion to overcome the difficulties of distortion and wasting of the combustion liner on the 18,000. What progress has been made along these lines and what are future indications?

2. Mention is made of improvements having been obtained in the life of the flame tubes of the distillate burning units 18,100 with new material. What new material is being referred to and along what lines is the development of new material proceeding?

3. In the long run, won't the economic possibilities of the gas turbine depend more on increasing the efficiency of the turbine rather than obtaining cheaper fuel?

4. What is the comparative fuel cost per 1000 ton-miles of the gas turbine to the diesel on the Union Pacific?

5. What is the comparative maintenance cost per 1000 ton-miles of he gas turbine to the diesel on the Union Pacific?

6. It is stated that, in the final analysis, possibly the only difference in operating costs may be the maintenance costs of the two types of power, and that this difference could only be between the gas turbine engine and the diesel engine since the transmission equipment is similar in both cases. Isn't maintenance a major factor?

7. Taking into account the different output characteristics of the two engines, is the transmission equipment that is most economical for the diesel engine also most economical for the gas turbine engine?

8. "It was also pointed out that in railroad application a good thermal efficiency would be required at 20 to 30% of the maximum output and that the efficiency at full load was of little significance." Has there been any development (such as along the lines of a control system) that would help to overcome this inherent difficulty?

9. "P. Draper of Shell amplified these comments by stating that, even if the locomotives were trouble-free, they could never make a really good case for gas turbines on railways because of their inherent design limitations." The so-called inherent design limitations are mainly connected with the fuel economy (especially critical at partial load) and the life of the turbine. Research is being conducted along such lines as will increase the fuel economy and increase the life. However, what thought has been given to the problem of the difficulty encountered by operating at partial load?

10. Why are they ordering 15 new units if maintenance troubles have been so frequent?

Author's Closure To Discussion

DR. MEYER informs us that the 2200-hp Swiss locomotive unit – which he designed – in test runs with a 220-ton trailing passenger train, consumed fuel at the rate of approximately 7 gal. per 1000 ton-miles. The same locomotive hauling a 110-ton train making frequent stops of considerable duration, consumed as high as 17.7 gal per 1000 ton-miles. This large range in fuel consumption as between a desirable train weight and a low train weight points up the low efficiency of the gas turbine locomotive at small loads.

He also points out that his company would be prepared to build a unit as large as 5000 hp, but that he would prefer to build units of 2500-hp sizes with one combustion chamber and a heat exchanger and with a maximum temperature not exceeding 1100 F. With this maximum temperature,

[a] See footnote 1 of main paper.

he feels that the units would be free from sulfur and vanadium effects caused when burning residual fuels.

The comments by Dr. Meyer are all very interesting and confirm the performance data on the Swiss locomotive unit No. 18,000 operating in England.

Mr. Hadley, who is a fuel expert, confines his remarks principally to the advantages of burning cheap residual fuel. In this connection I would recommend that his comments be studied by those interested in the gas turbine, especially in connection with fuel requirements.

As stated in my paper, it is desirable to employ the cheapest possible fuel with gas turbines, since they will consume probably over three times the fuel required for diesel power in normal operation. In this connection Mr. Hadley states that the present gas turbine fuel requirements are far from clear, although there is no question that it will be easier to burn cheaper fuels with turbines than with the diesel engines.

A panel discussion on residual fuels for the gas turbine, held during the Annual Meeting of the ASME in New York on Dec. 1, 1953, certainly indicates that the present gas turbine fuel requirements are very far from being settled.

As stated in my paper, it is my belief that the cost of residual fuel and distillate fuels will vary, according to the demand and the balance in production to suit the most economical processing of the crude oil. The demand today is for distillate fuel, and we may reasonably expect that most residual fuel will be processed accordingly.

In his closing statement Mr. Hadley refers to the adverse effect of elevation with the diesel engine. In actual practice, the commercial 4-stroke diesel with supercharger and the 2-stroke diesel with blower have excess air, which permits the development of the rated power up to approximately 8000-ft elevation. Above this elevation, the diesel locomotive engine must be derated.

Mr. Fahland unfortunately confined his comments to the question of appropriate locomotive unit size for Class I railroads rather than to diesel and gas turbine operating experience of the Union Pacific, in which we are all so interested.

Briefly, Mr. Fahland states that the 4500-hp unit employed on the Union Pacific is the most satisfactory size for American railroads. I stated in my paper that the indications were that 32,000 diesel locomotive units, giving a total of about 42,000,000 hp, should be able to operate Class I railroads adequately under war conditions. These units would average something less than 1500 hp per unit, if the units were 4500 hp each, as suggested by Mr. Fahland, the total investment in motive power would be approximately 145,000,000 hp instead of the 42,000,000 hp indicated.

One does not necessarily require a profound knowledge of railroad operation to appreciate the fundamental advantages of multiple-unit operation with standard-size units. It must be obvious that a switcher, a local freight locomotive, a medium-weight freight train locomotive, and a heavy-drag freight locomotive, together with the various sizes of passenger power, could not be very mechanically supplied with one 4500-hp unit. Herein lies the great superiority of the 1500-hp to 1700-hp diesel unit. The same type of unit can make up any size of locomotive requirement.

Mr. Fahland refers to jackknifing of four diesel units. I wonder if Mr. Fahland recognizes that these four units of presumably 6000 hp to 6800 hp have a hauling capacity of two of his 4500-hp gas turbine locomotives.

Mr. Fahland also refers to the difficulty of handling and turning the multiple-unit locomotive at terminals. A properly assembled multiple-unit locomotive should be capable of operating in two directions and there should be no wye movements or disconnection required at terminals. Actually, the record indicates that the Union Pacific makes very profitable use of the multiple-unit assembly in its operations, even on the Green River to Ogden runs.

We would, of course, like to know more details of the exact operating costs of the gas turbine locomotives on the Union Pacific. As indicated, very special operating conditions have succeeded in reducing the fuel consumption by at least 25% over what might be expected under normal run-of-the-mine conditions. We would like to know what the actual fuel costs are with the operating practices of the Union Pacific.

Mr. Barlow outlines a series of questions for which we would all like the answers. From the best analysis that I can make, railroads and manufacturers are not prepared to supply records of the maintenance cost for operations up to date. I might possibly exclude the Swiss Federal Railways from this category.

We must, however, recognize that, in the initial stages of operation, certain difficulties are bound to arise requiring redesign and special servicing, and to include these costs with any figures on maintenance might be quite misleading.

As stated in my paper, it would appear that some time must elapse before we can obtain satisfactory answers to the questions raised by Mr. Barlow. We may briefly reply to his questions as follows:

For answers to questions 1 and 2, this has been partly covered by Dymond.[a]

The answer to question 3 is "yes."

The answer to question 4 was given by Mr. Fahland in that fuel cost for the diesel is approximately the same as that for the gas turbine in the same service.

We have no answer to question 5 nor to question 10. They are definitely linked together.

The answer to question 6 is definitely "yes," and the answer to question 7 is also "yes."

The answer to question 8 is that no control system has been brought out to date specifically for the purpose of improving efficiency. The Brown Boveri Co. has designed its turbine to peak in efficiency at 70% load factor so that it gives a reasonable efficiency between 50 and 90% load. The Union Pacific gas turbine locomotives peak at 100% load factor, which does not help the fuel consumption at the lower load factor values.

Referring to question 9, Mr. Draper's full statement as published in the Dymond paper[a], indicates that the present locomotives in operation are simply in the early stages of the design and in time gas turbine locomotives will be built with considerable improvement in all respects as compared to those now in service.

I also received comments on my paper from F. Q. den Hollander, president of the Netherlands Railway, and Hedegaard Christensen, chief mechanical officer of the Danish State Railways. They both indicated that there is a great national interest by European railroads in the standardization of diesel locomotive units that had been developed in the United States. Their units would, of course, be generally smaller in size, but the indications are that the cost of electric transmission with diesel locomotives in Europe is abnormally high, due to the complete lack of standardization.

In concluding my comments on the discussion, I would like to point out the necessity of a better understanding by our railway mechanical officers of the important ratio of the permissible maximum speed to the permissible minimum speed with full horsepower output; in other words the flexibility of the transmission, whether it be electric or hydraulic. In this connection, I was interested to note, in a recent paper presented to the ASME on diesel locomotive hydraulic drives, that this ratio should be between 5 and 8. With electric locomotives proposed in this country and employed in Europe, no such exacting condition is specified.

Finally, it is evident that some time must elapse before the maintenance and regular fuel consumption values and cost are available with the gas turbine locomotives before we can properly evaluate their economic position in railway transportation.

Universal Positioning Seat Track

F. C. Matthaei, Jr., American Metal Products Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1954.

FOR many years, automotive engineers have designed automobile bodies around the physical characteristics of the average man. Such dimensions as eye level, wheel clearance, head room, and seating angle, therefore, did not accommodate those people other than average – as much as 60% of the American drivers. The successful development of practical, universal, and multipositioning seat tracks has now made it possible for the individual to tailorize the seating of his car to his own peculiar likes and requirements. This paper will be concerned with the physical variation of adult Americans, the automotive seating adjustments desirable to accommodate these variations, and the benefits resulting from such adjustments.

A recent paper[1] by E. C. Pickard has outlined the facts and figures used by Ford to arrive at "Oscar," their seating dummy, who represents the average American male. The development and existence of Oscar is a significant step forward and should be greatly appreciated by body engineers throughout the automotive industry. Oscar takes a lot of the guesswork out of our engineering problems. But let's take a look at the statistics from which Oscar was developed. You will recall that Oscar is 69½ in. tall, weights 165 lb, and represents the "80 percentile American male."

A study of reports of Government physical examinations, backed up by findings of privately conducted surveys,[2] reveals that approximately 53% of the male population is 2 in. shorter than Oscar; almost 35% is 3 in. or more shorter; while only 10% is 1 in. or more taller (Fig. 1). We must remember that Oscar represents the average man. This completely leaves out the women of the nation, bless them all, and this should not be done. Women are considerably shorter and lighter than men. You will note on the chart that very few women are as tall as Oscar, while 80% are shorter than Oscar by some 4 in. or more. Fifty per cent are 6 in. or more shorter.

Just as Oscar is taller than a large percentage of

[1] "Use of 'Oscar' in Seat Design," by E. C. Pickard. Presented at SAE Annual Meeting, Detroit, Jan 11, 1954.

[2] Department of the Army, Office of the Surgeon General, January, 1953; "Survey in Seating," by E. A. Hooton, Harvard University, 1945; "Dimensions of Average Driver," by W. E. Lay and E. R. Gooding, University of Michigan, 1948.

THE universal positioning seat track is said to be the answer to the problem of providing suitable automotive seating for practically all drivers. It allows each driver to adjust the car seat to the position that best satisfies him.

The author points out that providing these adjustments gives a number of advantages, such as:

1. The visibility of the driver can be at a maximum.

2. Each driver of a car can adjust the seat to suit his own individual size – and taste.

3. Less driver fatigue.

The Author

F. C. MATTHAEI, JR. (J '50) is sales engineer for the American Metal Products Co. He attended the University of Michigan and received a B.S. degree in mechanical engineering. Mr. Matthaei has been continuously associated with American Metal Products since 1947.

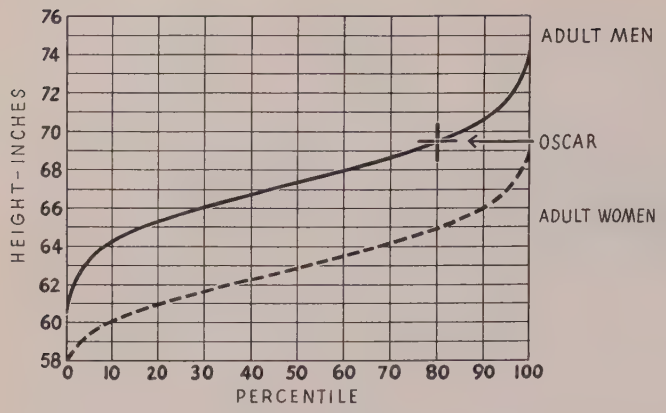

Fig. 1 – Height percentiles – American adults

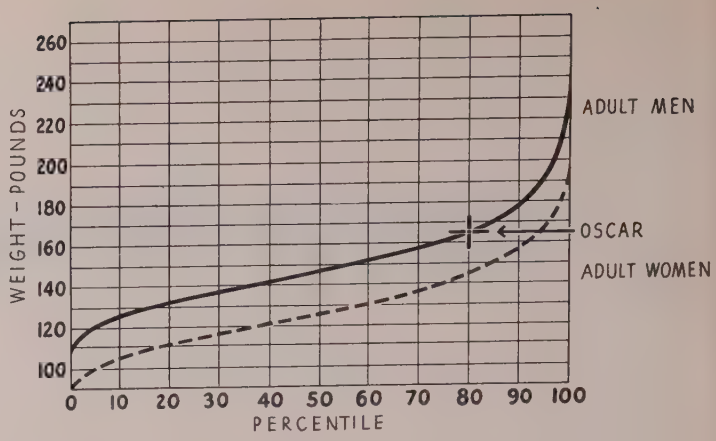

Fig. 2 – Weight percentiles – American adults

American adults, he is also heavier (Fig. 2). Oscar weighs 165 lb; yet 60% of the male population weighs 160 lb or less, and 30% weighs 138 lb or less. Roughly 15% weighs 170 lb or more. Women also weigh less, with 60% being lighter by 35 lb or more.

You can readily see, therefore, that, although Oscar is a good man to work with, he is different dimensionally from most of us. In fact, if we use the combined figures quoted before as typical of all our licensed drivers (70% men, 30% women[3]) 53% of the American drivers are 3 in. or more shorter than Oscar, and 64% are 2 in. or more shorter than Oscar (Fig. 3).

So much for the height and weight of the human populace. It is interesting to note other variations of the human body that affect seating (Fig. 4).

As you may or may not know, the length of your leg from the bottom of your heel to the top of your knee varies as much as 4½ in., in an average 22-in. dimension. Eighty per cent of the males have a 2⅝-in. variation over this dimension. The dimension, taken while you're sitting down, from the seat of your pants to the top of your head (an average of some 36½ in.) will vary 5¼ in. Eighty per cent of the population is within a 3-in. variation. Even so small a dimension as the height from the centerline of the eye to the top of the head will range from 3 in. to 5¾ in., with 80% of us varying between 4 and 5 in.[4]

It is quite evident, therefore, that, although Oscar represents as accurate a picture of the average American male as possible, more than 60% of the American drivers vary sufficiently from Oscar's dimensions to warrant some adjustability in seating.

Then, too, there are other variations that enter into the individual seating picture – they are the personal likes and dislikes of each one of us. By that I mean, some people like to sit erect, others slouch. Some of us put pillows behind our backs,

[3] P. 36 of "Automotive Facts and Figures," 1953 edition. Pub. by AMA, Detroit.

[4] Department of the Army, Office of the Surgeon General.

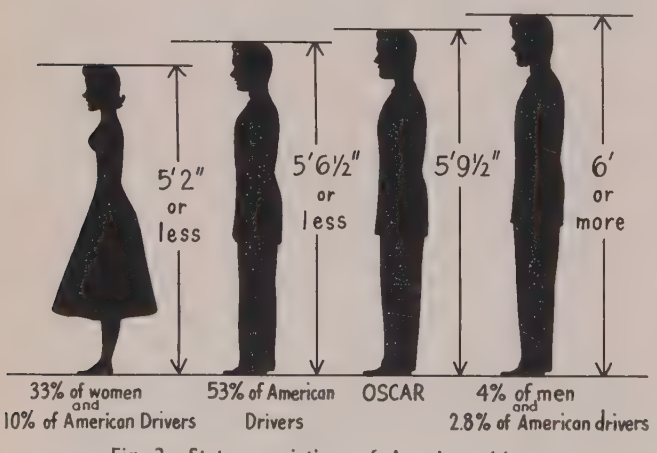

Fig. 3 – Stature variations of American drivers

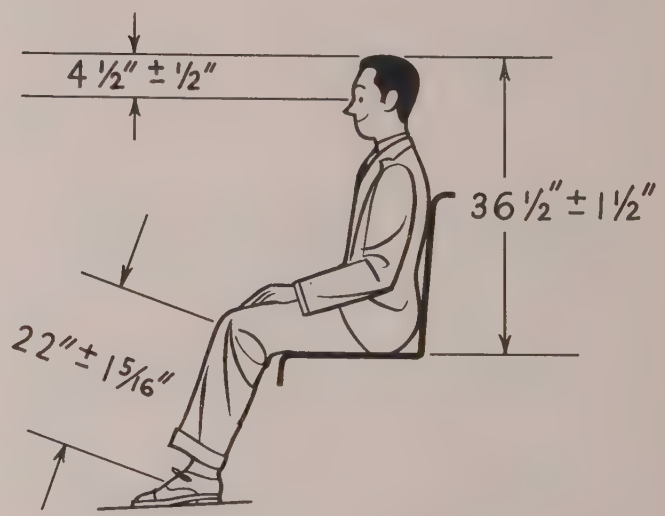

Fig. 4 – Dimensions and variations representing adult American males

while others would like to put pillows under our knees.

The present fore-and-aft inclined track that rises as it comes forward takes care of some of those variations but, again, a track that is universally adjustable would satisfy a greater percentage of the individuals.

After studying statistics similar to those presented here, the definite need for a universal positioning seat track becomes obvious. As a result of such a study, our company embarked on the development of a multipositioning seat track. We were not sure as to the range of vertical and angular adjustment required to accommodate the maximum percentage of all drivers. We, therefore, built a track that was adjustable manually for three different conditions:

1. The front of the seat could be raised or lowered individually within 1½ in. of travel, in ⅛-in. increments.

2. The rear of the seat could be raised or lowered independently within 1¾ in. of travel, also in ⅛-in. increments.

3. By combining the vertical movement of the front and rear adjustments, the angle of the seat assembly could be varied ± 7 deg from normal.

We installed the track in a car and asked a number of people to adjust the seat to give them the most comfortable driving position. The results of this survey were recorded. They revealed that:

1. The individual's personal likes and dislikes have a greater bearing on his seating position than do his physical dimensions. In other words, people of the same height and weight would adjust the track into widely different positions.

2. The seating angle afforded by the present production automobile seats and fore-and-aft seat tracks represented the average angle chosen as the most comfortable – approximately 15 deg (the angle of the seat cushion to a horizontal line). The range of angle selected as most comfortable varied, however, from 11 to 17 deg.

3. Approximately 75% of the people surveyed positioned the seat higher than is afforded by the present fore-and-aft track. This vertical adjustment ranged through the full movement available.

As a result of this survey, our present tracks were designed with 1½-in. vertical adjustment in the front, and 1¾ in. in the rear. Some automobile manufacturers thought the vertical adjustment most necessary, and are now using tracks movable vertically and horizontally by power without angular adjustment. Another manufacturer firmly believes that the angular adjustment is all important and is now using a track manually adjustable vertically and angularly as well as horizontally.

We have now covered the range of variations in the human body and the amount of seat track adjustment required to satisfy this variation. What are the benefits obtainable from these adjustments?

The most obvious and all inclusive benefit is

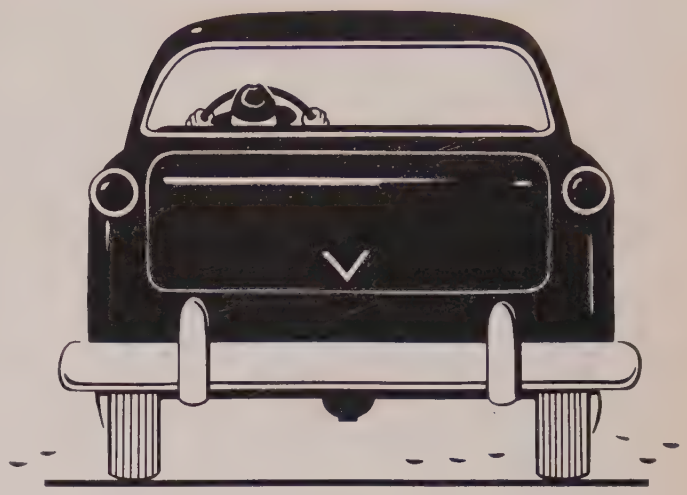

Fig. 5 – Vanishing phenomenon of American road

Fig. 6 – Visibility – without vertical elevating seat track

Fig. 7 – Visibility – with vertical elevating seat track

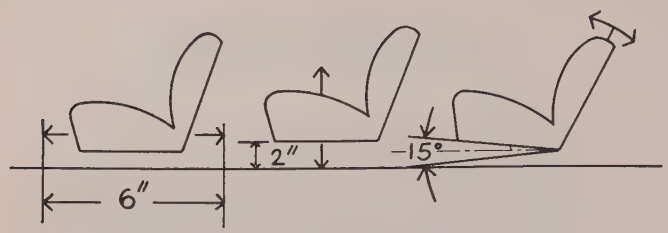

Fig. 8 – Movements available – automotive seat tracks of future

that, with a universal positioning seat track, each individual becomes his own seating engineer. He can and will adjust the seat to the position that most satisfies him. The ramifications of this are many:

First, the driver's visibility (all important for safety) can be at a maximum. We can forget the shims under the tracks, the pillows under the driver, and behind his back. With a vertically adjustable track, a phenomenon of the American road will pass into history – the apparently driverless car (Fig. 5). That short lady or gentleman can now be elevated above the steering wheel. There are truly new horizons ahead for these individuals.

Then, too, there's another phase of improved visibility. How many of you, satisfied with your seat angle for normal driving, have driven home on a dark, foggy night sitting well forward on your seat peering anxiously over the wheel. With the universal positioning track, you merely set the seat in that position and arrive at your destination (I hope safely) without the strained back and neck muscles.

There is still another phase of visibility that will interest most of us, and I'll touch upon that briefly. In Fig. 6, we have visibility *without* the vertical-elevating seat. You can see that *with* a vertical-elevating seat track (Fig. 7) visibility is much improved.

As the second benefit of the universal or multi-positioning seat track, not only is the one driver of the car able to select his best seating position – all drivers of the car can make this selection. This brings the ladies back into the picture. As many recent surveys point out, they are an important group to satisfy. The high heels most women wear do not help much for height when they are seated. Women appreciate the vertically adjustable track.

A third important benefit of the universal positioning seat track is a lessening of driver fatigue. By adjusting the seat to meet the individual's comfort requirements, he does not tire as easily. He can support his shoulders as much as desired by changing the back angle. The long-legged fellow can support his knees by tipping the seat back; the short-legged man can relieve pressure under his knees by tipping the seat forward. In driving over long distances, a small change of one or two notches, changing the seat angle, will shift the center of weight of your body on the cushion enough so that those fatigued sore spots do not occur.

These are broadly the benefits obtainable by use of a universal positioning seat track. Besides creating a more comfortable driver, we have a better driver in that he has the optimum visibility obtainable and is less apt to suffer from fatigue.

When we get into a discussion of what the future holds as a result of the universal positioning seat track, a broad vista opens. All body engineers are concerned with various dimensions such as head room, eye to header, eye to cowl, the clearance between the seat contours and the steering wheel, both horizontally and vertically, and others of that nature. Oscar was created to help with these dimensions. If the universal positioning seat track is used only as an accessory, the stylist and the body engineer will still be limited by these dimensions. If, however, the track became a standard item, its adjustability would permit the stylist and the body engineer a greater range of design. With the present trend to longer, lower cars, this wider range of design should be of considerable help.

In the last decade, automotive engineers have brought an amazing list of engineering accomplishments into common usage – the automatic transmission, power steering, and power brakes (to name a few). The universal positioning seat track enables the body engineer to keep pace with these improvements.

The track of the future will be truly a 6-way track, adjustable fore and aft horizontally, up and down vertically, and inclined forward or back angularly (Fig. 8). These adjustments will be made manually for low-priced automobiles and by power in the more luxurious fields. The vertical movement available will be between 2 and 2½ in., compared with the present 1½ and 1¾ in. The angular adjustment will be ± 7 deg from the normal seating angle.

In conclusion, the physical variations of the human body, together with the whims and fancies of each of us as to seating position, make the universal positioning seat track a "must" for cars of the future. Besides opening new fields in design for the body engineer, this type of seat track will provide each individual with a more comfortable, safer ride. There is a great future for the universal positioning seat in the automotive world.

Acknowledgments

The author gratefully acknowledges the assistance of E. C. Pickard and E. A. Langtry of the Ford Motor Co., and Prof. W. E. Lay of the University of Michigan, who furnished valuable data concerning the American population; as well as R. J. Williams, R. E. Heyl, and all the other members of the product development department of American Metal Products Co., whose engineering accomplishments made this paper possible.

Human Engineering:

A New Approach to Driver Efficiency and Transport Safety

Ross A. McFarland, Harvard School of Public Health

This paper was presented at the SAE National Transportation Meeting, Chicago, Nov. 2, 1953.

IN recent years a new discipline has been developing in which the engineering and biological sciences are being integrated. This field of study is commonly referred to as human engineering. One of its primary objectives is to improve safety by designing equipment in terms of human capabilities and limitations. Mechanical design must be intimately related to the biological and psychological characteristics of the operator. It seems reasonable to require that machines be designed from the man outward, with the instruments and controls considered as extensions of his nervous system and body appendages. This implies that machines and working areas must be built around the operators rather than placing them in a setting without due regard for their requirements and capacities. Unless this is done, it is hardly fair to attribute so many accidents to human failure as is usually the case.

If this point of view is carried out in practice, fewer accidents should result, training costs should

HIGHWAY safety – an important problem in the truck and bus industries – is governed by many factors, one of the most important being the human element. This paper describes the Harvard School of Public Health research program concerning the human factor in vehicle design and operation.

The project is sponsored by the Commission on Accidental Trauma of the Armed Forces Epidemiological Board, Department of Defense; the National Association of Automotive Mutual Insurance Companies; the American Trucking Associations, Inc.; and the National Association of Motor Bus Operators.

Some items of this investigation include the study of near accidents; detection of the accident repeater; personal adjustment and morale of drivers; human body size and capabilities with relation to controls, instruments, seats, and vision; the effects of temperature, ventilation, noise, and vibration; and desirable safety features to protect the operator if an accident occurs.

The Author

ROSS A. McFARLAND (M '52) has been associate professor of industrial hygiene at the Harvard School of Public Health since 1948. He graduated from the University of Michigan and received his doctorate in physiological psychology at Harvard in 1928. Dr. McFarland taught at Columbia University for eight years and then from 1937 to 1947 devoted his entire time to research at the Harvard Fatigue Laboratory and the Division of Research at the Harvard Business School.

be reduced, and extensive redesign of equipment after it is put into use eliminated. This principle holds true whether an employee is operating motorized equipment on the highway, flying an airplane, operating a lathe or punch press, or carrying out any of the numerous mechanized duties of a manufacturing or base repair shop. In this regard, two publications by the author, although dealing primarily with aircraft, nevertheless have broad implications for the application of human engineering in all fields.[1, 2]

More precisely stated, first, the engineer must have an understanding of the sense organs and the characteristics of human perception if satisfactory information is to be supplied from sounds and lights in the operation of equipment. In general, man is a poor monitor of equipment. Secondly, the engineer must not place unreasonable demands upon the pilot in regard to the physical responses involved, that is, in reaching or operating a control or lever or in clearly differentiating between them. Otherwise, an inadvertent operation of a switch or control may occur. Third, the engineer must consider that the operator of equipment is frequently suffering from temporary impairment of functions causing inefficiency, such as fatigue, and there may be changes in function as a result of aging. Finally, the physiological and environmental conditions such as temperature, humidity, toxic gases, and other factors must be considered, or physiological limits may be exceeded.

A. *Major Areas for Research in Human Engineering* – The major areas in which the biological and engineering sciences can collaborate most effectively during the next decade are as follows:

1. Operational job analyses, including studies of critical incidents and near accidents.

2. Layout of the working space, including variations in human body size for ease and efficiency of operation.

3. Design and arrangement of displays to provide information for operating under normal and emergency situations.

4. Design of controls and their arrangement.

5. Systems analyses including link values and intercommunications.

6. Temporary characteristics of the individual, such as fatigue, loss of sleep, diet, alcohol, drugs, learning.

7. Changes in function as a result of aging.

8. Control of environmental factors influencing physiological and psychological reactions.

9. Prediction of error and provision for margins of safety.

A comprehensive report of our three-year study[3] is now available and covers all of the above areas in detail.

B. *The High Incidence of Accidents on the Highway* – The high incidence of temporary and fatal injuries resulting from all types of accidents constitutes a major problem in both civilian and military populations. The loss of life and incapacity resulting from this cause is greater than from any known disease entity. In addition to the large number of fatalities, accidents also give rise to a tremendous number of temporary and permanent injuries, damaged equipment, losses in productive output, and extensive use of medical personnel and facilities.

Accidental fatalities rank first among the 10 leading causes of death for persons in the age range from 1 to 44 years. The fact that these victims are in the younger age groups, in contrast to those whose deaths are attributable to certain degenerative diseases, implies an enormous cost to the productive resources of the country in terms of life-years. Measurement of nothing but the fact that a death has occurred is less meaningful than the measurement of how much life-time has has been lost prematurely. An 80-year-old man dying of heart disease and a 12-year-old boy run over by a car are each counted as one death under the presently used methods.

In civilian life, highway accidents are assuming even greater importance with the increasing volume and speed of such equipment. Each year in the United States there are approximately 40,000 motor vehicle deaths, 100,000 permanent injuries, and about 1,000,000 temporary total disabilities resulting from this source. Due to the increased mileage and exposure, the actual number of deaths and injuries has increased to such an extent that 10% of the population of the United States may be expected to be killed or injured in highway accidents in a period of about 15 years. In fact highway accidents have increased to such an extent that almost everyone in the population will be influenced at some time or another during the course of their lives. The importance of this situation has led us to compile a series of over 1000 abstracts of pertinent articles covering all aspects of highway safety.[4]

Preventive Medicine, Public Health, and Safety

The control of accidents falls within the province

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[1] See "Human Factors in Air Transportation: Occupational Health and Safety," by R. A. McFarland. Published by McGraw-Hill Book Co., Inc., New York, 1953.

[2] See "Human Factors in Air Transport Design," by R. A. McFarland. Published by McGraw-Hill Book Co., Inc., New York, 1946.

[3] See "Human Factors in Highway Transport Safety," by R. A. McFarland and A. L. Moseley. Published by Harvard School of Public Health, Boston, 1954.

[4] See "Human Variables in Vehicular Accidents: A Review of the Literature," by R. A. McFarland, R. C. Moore, and A. B. Warren. Published by Harvard School of Public Health, Boston, 1954.

of preventive medicine and public health because of the important role played by human variables. It is not generally appreciated that injuries, as distinguished from disease, are also amenable to the epidemiological approach, and that accidents follow some of the same biological laws as do disease processes. In most instances, there is multiple causation, and attempts at control should involve consideration of the interaction of agent, host, and environment, in other words, the vehicle, the driver, and the highway. Although the host is of primary medical concern, the agent and the environment must also be considered in effective preventive measures. This applies equally to military and civilian populations, under all conditions, including combat.

Attempts should be made first to set forth the basic physical, physiological, and psychological characteristics of the host or driver. When such facts are associated with the agent under given environmental conditions and at specific times and places, information can be obtained so as to understand and prevent accidents. Factual information of this type can be discovered only by carefully controlled experimental studies, epidemiological surveys, and statistical analysis. A dynamic and continuous safety program can produce positive results only by the constant application of the fundamental principles resulting from such studies.

A. *Promising Approaches to the Prevention of Accidents* – 1. Analysis of Activities and Jobs – One of the first steps in assuring safety on the job is to make an analysis of what is required of the individual and to study the more critical features of each job. Many accident potentials can be revealed by this procedure. Each type of equipment and its operation must be subjected to thorough study and interpretation, in order to understand what is expected of employees. Techniques involving near-accidents are often revealing, and minor incidents should be recorded and interpreted. The critical-incident technique offers promise in understanding and preventing serious accidents.

2. Health Maintenance Examinations – Much remains to be done in the highway transport industries in applying modern conceptions of preventive medicine and occupational health in safety programs. The contribution of the medical officer in preventing accidents can be effective. Interval health maintenance examinations can aid in preventing errors or accidents which result from incipient or actual illness. The physician can aid greatly in indoctrinating personnel in regard to accident prevention at the time of the examination or during the treatment of injuries.

3. Detection of Accident Repeaters – An attempt should be made to determine whether some individuals at each establishment appear to be more liable or "prone" to accidents than others and have higher accident rates. Studies of accident repeaters involve (a) the determination of whether there are important differences between individuals in their accident liabilities and (b) if differences do exist, devising methods for detecting those individuals who are especially liable to accidents. Those individuals having repeated accidents are often impulsive, aggressive, and resentful of their superiors. Such persons are also frequently known to various social agencies and also to the police courts.

4. Personal Adjustments – A very promising field in regard to accident causation relates to the role of mental and personal adjustments. Maladjustments and emotional conflicts are often precipitating factors. The influence of worry and apprehension are undoubtedly important in the causation of fatigue and operational failures. Rest and sleep are important, and the excessive use of alcohol is known to be an important contributing factor to accidents, especially while off duty.

5. Morale – There is general agreement that poor morale figures prominently in the causation of accidents. Interest in the operations, adjustment to associates, and attitude toward superiors are believed to be important.

6. Supervision and Leadership – Finally, the role of leadership and supervision is significant. Several recent studies indicate that high accident rates tend to follow some supervisors wherever they are located, even if they are transferred from one place to another.

Human Engineering with Special Reference to Vehicle Design

Another approach relates to the effective integration of the worker and his equipment. Human body size and capabilities should be considered in the design of instruments, vehicles, and working areas. Where this principle is ignored, situations may develop in extreme cases where only 5% of the operators can reach and comfortably operate the hand brake. In others only 60% are accommodated with regard to knee height between the pedals and the steering wheel, while one vehicle accommodates only the smallest 40% with reference to the general body size.

A. *Human Sizing Problems* – In order to improve the ease, efficiency, and safety of operating all types of equipment, consideration should be given (1) to the application of anthropometric (human body size) data to design and (2) to its use in a dynamic setting. In many instances, an engineer is given instructions to design the equipment and the operator's area in such a way as to "prevent reaching beyond range of vision," "avoid off-balance positions," and "provide adequate headroom and clearance." He is rarely supplied with quantitative descriptions to meet these requirements. On the

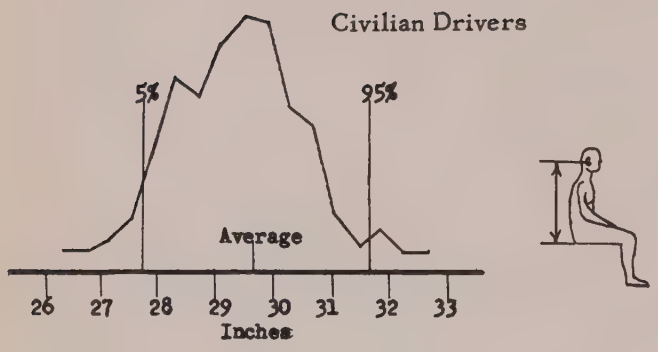
Table 1 – Normal Sitting Eye Height (Distribution in Percentiles)

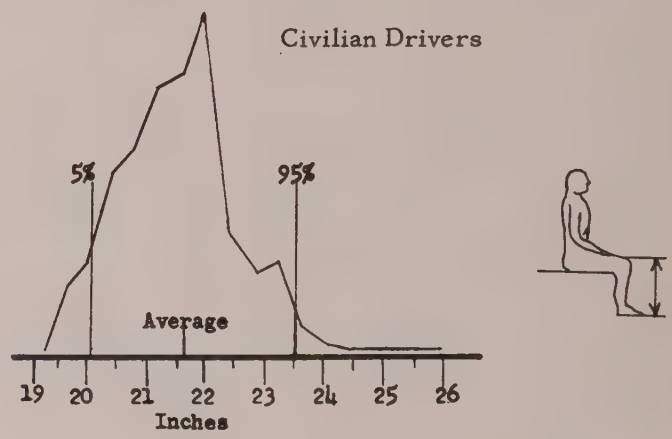
Percentiles	Sitting Eye Height for Civilian Drivers, in.
5	27.72
10	28.11
15	28.39
20	28.58
25	28.74
30	28.98
35	29.13
40	29.29
45	29.45
50	29.61
55	29.72
60	29.88
65	30.00
70	30.16
75	30.43
80	30.55
85	30.75
90	30.98
95	31.61

Number: 309
Range: 26.38 – 32.68
Median: 29.61

Table 2 – Knee Height (Distribution in Percentiles)

Percentiles	Knee Height, in. Civilian Drivers	Army Drivers
5	20.08	19.78
10	20.43	20.23
15	20.63	20.51
20	20.83	20.69
25	20.98	20.87
30	21.18	21.02
35	21.30	21.16
40	21.42	21.30
45	21.58	21.43
50	21.69	21.56
55	21.81	21.68
60	21.93	21.82
65	22.05	21.96
70	22.16	22.10
75	22.24	22.26
80	22.36	22.42
85	22.56	22.67
90	22.84	22.96
95	23.50	23.42
	Number: 301 Range: 19.29 – 25.98 Median: 21.69	Number: 2376 Range: 17.72 – 25.59 Median: 21.56





other hand, if quantitative measurements are provided, the "average-value" fallacy is often committed. Although such values can be applied to special cases in the design of equipment, arrangements based on averages would be unsuitable for 50% of the operators in a normally distributed group. Provision should be made for 90 or 95% or any other predetermined percentage of the potential operators. If this is not done for such a requirement as arm reach for the operation of manual controls, many of those who exceed or are below the average valves would be handicapped. Standard sources of anthropometric data relating to body measurements are available.

Instead of averages, percentile distributions tables are used which show the measurements for any given human dimension in tabular form for each 5% of the population, from the smallest or 5th percentile to the largest or 95th percentile. Thus, by choosing the 5th percentile as the minimum and the 95th percentile as the maximum a vehicle may be designed to fit fully 90% of the population, a desirable practice. To enable such a practice to be carried out, the Harvard School of Public Health has taken 31 functional measurements on 360 professional drivers. The percentile

distribution tables are now available for this series.[5] In addition this manual contains a review of established human engineering procedures and attempts to outline methods applicable to truck and bus design.

B. *Applying the Data* – The problem of integrating anthropometric data with the static dimensions of the cab is one of varying complexity. In some cases, integration is relatively simple. For example, it is possible to determine the minimum width of the seat merely by taking the widest "seat breadth" in the distribution of drivers and making an added allowance for comfort. This can be done because a man's buttocks are relatively static while driving. Similarly, it is possible to determine what should be the range of vertical adjustment in a seat to maintain a uniform optimum eye level for drivers by subtracting the seated eye level of the shortest 5% in the distribution from the tallest 95%. This sets the range of vertical adjustability for this purpose at 4 in. (Table 1), and it will permit the driver to maintain an adequate eye-level position for any given horizontal adjustment. The latter, in turn, may be determined on the basis of several factors such as the abdomen depth of the drivers and leg length for operation of foot controls. If only the average eye-level value is used, it may be 2 in. too high for the smaller drivers and 2 in. too short for the taller ones. Again, if it is known that the 95th percentile for drivers knee height is 23½ in. (Table 2), to accommodate all but the largest 5% of the drivers, there must be at least 24½ in. (23½ in. + 1 in. shoe thickness) between the top of the brake pedal and the bottom of the steering. Otherwise the situation illustrated in Fig. 1 will arise. In the vehicle shown in Fig. 1, the distance between the top of the brake pedal and the bottom of the steering wheel permitted the driver to raise his foot only ½ in. off the pedal, since at that point his leg was pressing into the wheel (left). The driver then moved his heel over to the left, but when he tried to slide the top of his shoe onto the brake, the sole caught on the pedal (right). In some vehicles, the alternative is to move the leg out to the right of the steering wheel, apply pressure to the brake with the leg at an angle, and then straighten the leg as the pedal is depressed. The loss of valuable time in this operation is evident. Moreover, in one particular vehicle, the gearshift lever in the left positions was too close to the steering wheel to permit the leg to slip in between.

Fig. 1 – Human sizing difficulties in the operation of the foot brake

In other cases however, the problems are more complex, for, while a man is driving a vehicle, he is in a dynamic situation. As a result, static human dimensions cannot always be applied to the dynamic situation without additional study. However, mockups can be built to aid in determining the optimum placement and design of the various controls and seats in both static and dynamic situations.

For this purpose a new test seat has been built for the Harvard School of Public Health to circumvent the practical difficulty of testing current model vehicles in the laboratory against variation in driver size. A wide range for all horizontal, ver-

[5] See "Human Body Size and Capabilities in the Design and Operation of Vehicular Equipment," by R. A. McFarland, A. Damon, H. W. Stoudt, and others. Published by Harvard School of Public Health, Boston, 1953.

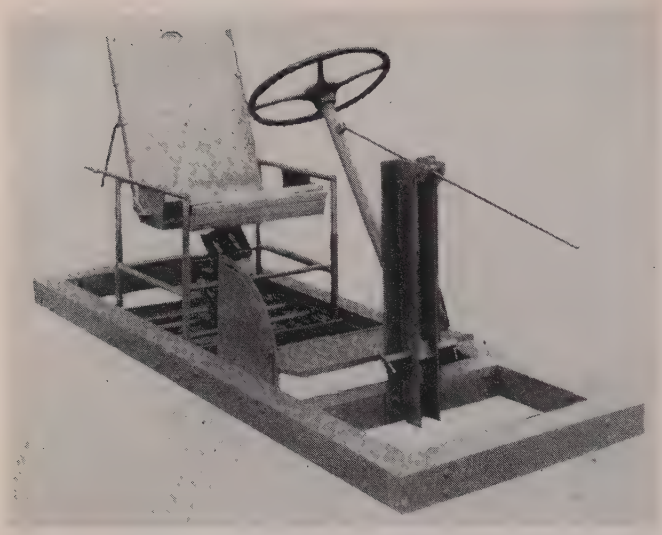


Fig. 2 – A preliminary view of the vehicle test seat constructed by the Harvard School of Public Health to determine optimum design and placement of the various controls in relation to driver's seated position

tical, and angular adjustments is provided for. With the completed equipment it is possible to reproduce the essential components of any motor vehicle in the laboratory and thus to test it adequacy in fitting the men who must operate it. Fig. 2 shows a preliminary view of this test seat. Another phase is anticipated where the components of the device will be adjusted in terms of the range of human size in order to find the optimum compromise arrangements. It will also be possible to determine the dependence of one functional variable upon another, thus permitting the establishment of an order of importance of the dimensions and angulations involved. Such variables include seat adjustment, pedal location, steering-wheel dimensions, and the like. Inherent in this approach is the concept of the dynamic nature of the job. The use of jointed full-scale manikins was considered and discarded in view of the dynamic requirements for adequate testing. Living manikins who have been selected as approximating the extremes (or 5th and 95th percentiles) and average (or 50th percentile) of the various important body dimensions will be used. The preliminary dimensions established will then be tested under simulated driving conditions over a period of several hours. Objective measurements, such as reaction time and the field of vision, will be taken in addition to such subjective material as the drivers own estimation of the equipment, his fatigue, comfort, and so forth.

C. *Designing to Suit the Operator* – In addition to problems of body size, many illustrations may be given of the way in which modern technology has produced machines without adequate concern for the capacities of the operators. If significant progress is to be made in the future, mechanical design must be related more directly to the psychological and social habits of man.

An illustration of this situation may be drawn from the confusion of controls in operating the flaps and landing gear of aircraft. The controls are often too close together or their position is reversed in different models. Thus, inattentive manipulation or mistaken identity resulted in 457 aircraft accidents in the U. S. Air Force during the 22-month period from January, 1943, to November, 1944. Of these accidents, 273 occurred in advanced trainers and 184 in fighters, bombers, and transports. This same error occurred in 1948 on a large commercial air transport. The copilot had recently been transferred from DC-4 to Constellation operations. After making a landing at Shannon, the captain asked the copilot to raise the flaps. The switches were exactly reversed in location on the two types of planes, and the safety mechanism was defective. As a result, the landing gear was retracted and the airplane broken in half. Recently an outstanding NACA test pilot told me that he had committed the same error after being transferred from one type of plane to another.

Returning to land vehicles, we see a similar example where engineers, in attempting to compensate for or eliminate low levels of mechanical skill in passenger-car operation and to reduce the work load of shifting in trucks up to the 1½-ton class, have provided the automatic transmission. The passenger cars of our manufacture have, however, a defect which is related to engineering on the one hand and the habits of drivers of conventional shift automobiles. The engineering defect makes it possible to shift the vehicle into reverse gear while the car is travelling at fairly high speeds. Since the physical forces necessary to accomplish this transmission change is measured in a few ounces, the defect has potentially serious consequences.

D. *Safety Factors in Designing for Sudden Deceleration* – The ability of the human body to withstand sudden deceleration and impact forces is extremely important for the operator's safety in case of collision. Research along those lines by Stapp and DeHaven has indicated that the force of many accidents that are now fatal is well within the physiological limits of survival. Many needless injuries, both fatal and serious, are caused by the unfortunate design and placement of certain objects and structures.

It was found that a human being can withstand more than 100 *g* and be stopped from a velocity of 50 mph within a distance of 6 in. without injury if the force of contact is sufficiently spread out. On the other hand, a speed of 15 mph can cause death if the momentum of the head is not checked during rapid deceleration of surrounding structures. If a 10-lb object, the approximate weight of a human skull, falls 1 ft and strikes an area 1 in. square the deformation would be very slight. On the other hand, under the same conditions, if an object 1 cm

square is struck, a puncture fracture would result.

Especially pertinent to vehicle operation is the fact that if an individual in a motor vehicle is properly suspended, such as by a shoulder harness, he should be able to survive uninjured a head-on collision with a brick wall, in which the vehicle stops in 2 ft from a contact velocity of 57.4 mph.[1]

Auto crash injury research has been undertaken at Cornell University Medical School as an extension of a similar program in aviation. Automobile crashes are analyzed so that structures repeatedly causing injuries can be identified. The objective is to obtain accident and correlated injury data whereby safety groups can evaluate safety needs and engineers can undertake modifications to minimize injury in future accidents. These will always occur, since such causes of accidents as human carelessness, fatigue, recklessness, stupidity, and intoxication can never be entirely eliminated.

The first phase of this program is a pilot study based on cases reported by the Indiana State Police, in which the State Troopers fill out the accident reports, take photographs, and provide physicians and coroners with medical reports. This information is forwarded to Cornell, where the individual accidents are analyzed and complete records made for cross reference to similar cases. In this manner it is hoped to determine some of the specific features of automotive design that cause unnecessary injury in crashes. The industry, to give a single example, is well aware of the problem of doors popping open in collisions. When it is possible to get accident data on cars in which the doors pop open frequently and on other cars in which the doors remain closed even on severe impact, it will be possible to effect redesign throughout the industry. The eventual results of this study should be most valuable to all those concerned with the truck and bus industries.[6]

Additional Physical Variables Influencing Safety and Comfort

An additional factor influencing the efficiency and safety of the operator is the physical variables of his environment. In vehicle operation these may be broken down into temperature, humidity, and ventilation, noise, and vibration. Extremes of any of the factors will result in the discomfort and lowered efficiency of the driver as well as the discontent of the passenger. For each of these variables, there are reasonably well defined ranges of comfort and discomfort.

A. *Temperature, Humidity, and Ventilation* – Since a change in any one of these variables will effect the other two insofar as they determine comfort, they must be considered as a unit. A fairly wide range of temperature can be considered as

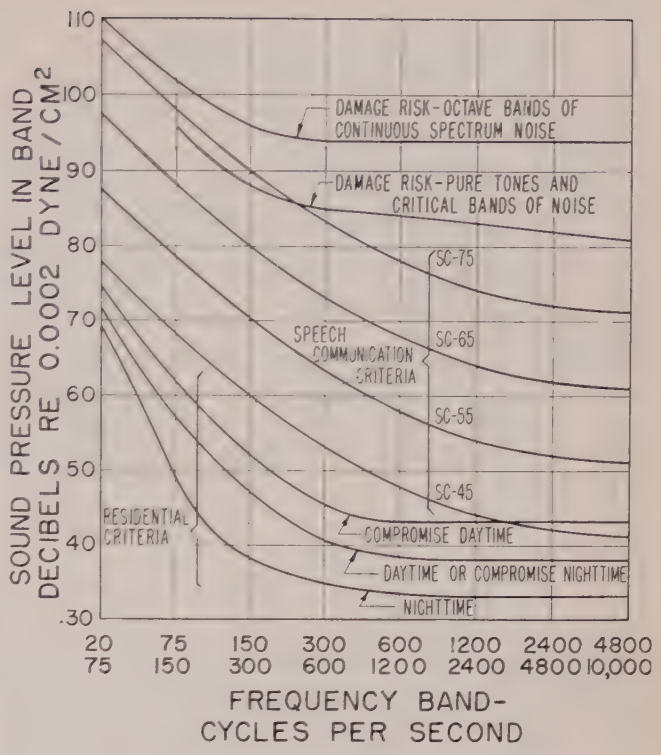


Fig. 3 – Tentative criteria for noise control showing the various levels of speech communication criteria

stimulating if it is related to low humidities. However, once the latter falls below 15% an uncomfortable drying out of the eyes and more will result. In general it may be said that temperature of 68 to 72 F should be maintained (if dressed in everyday clothing) while a humidity range of 25 to 50% is quite acceptable.

One of the most important requirements for passenger comfort relates to air movement and the amount of fresh air that must be used in ventilating the cabin. Very low rates of air movement can produce a feeling of stuffiness. Under such conditions, body odors and tobacco smoke become very irritating. There should be a well-diffused sectional distribution of air in buses to prevent channeling or seat to seat accumulation of odors. Optimally 35 to 40 cfm of recirculated air should be supplied per passenger, of which 20 cfm is fresh air. Air velocities should be between 20 and 60 fpm in order to maintain a sense of freshness without creating undesirable drafts.[1, 2]

B. *Noise and Vibration* – It is now well recognized that exposure to noise of sufficiently great intensity and duration will produce varying degrees of hearing loss, impair the efficiency of the worker, and possibly give rise to accidents. The noise levels of seven current truck models tested by the Harvard School of Public Health averaged 69 db (a measure of relative sound intensity). The range in one truck model was 82 to 89 db. These tests were taken in a closed cab with the motor

[6] See "Notes on Planning Conference," Auto Crash Injury Research. Published by Cornell University Medical College, New York, 1952.

Table 3 – Limits for Variables Influencing Driver and Passenger Comfort

Variable	Desirable	Maximum or Minimum Value for Comfort
Ventilation	Sufficient additional fresh or recirculated odor-free air to remove all odors	20 cfm of fresh air
Air temperature	Ambient temperature adjusted to give effective temperature of 63 – 71 ET (winter) or 66 – 75 ET (summer)	70 ± 2F dry bulb
Air velocity	Adjusted to ambient temperature to give appropriate cooling power without drafts	40 – 60 fpm
Humidity	Adjusted to air temperature and velocity to maintain comfort	25 – 50% relative humidity
Carbon monoxide	None	0.005%
Carbon dioxide	Not a requirement as long as ventilating requirements for removal of odors are met	0.5 – 1.0%
Noise	Reduced to permit conversation at at least 3 ft with no extra effort	75 db overall, and 50 – 60 db in 1200 – 2400 band
Vibration	Reduced to below threshold of perception	0.008 in. at 8 cps or the equivalent

idling, not under highway conditions which would tend to increase the readings.

Fig. 3 shows tentative criteria for speech control which can be correlated with the above findings.[7] At the 55-db level (SC 55 on the chart) conversation is intelligible at a distance of several feet with a normal voice. At the 65-db level, however, communication is difficult, while at 75 db communication is marginal. All but two of the seven trucks tested exceeded the 75-db marginal speech level, thus indicating the necessity for further noise reduction in vehicles of this nature.

It will be noted that the speech communication level expressed in decibels varies with the frequency, or "pitch," of the sound. At low frequencies a given background noise is less detrimental to communication than it is in higher frequencies. For the most part the noise produced by trucks and buses are in the lower frequency range which tends to alleviate, but by no means dispel, the difficulties mentioned above. A reduction in the intensity of any frequencies above 600 cps (engine noise is below this figure) will result in an improved intelligibility of speech and increased comfort, even though the overall noise level remains the same.

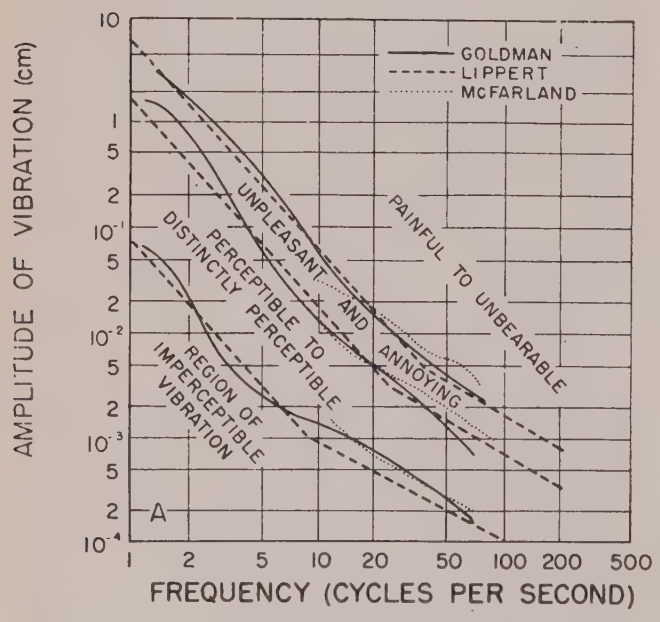


Fig. 4 – Subjective response to vibration in terms of frequency and displacement amplitude as determined by three investigators[1]

It is apparent that for safe, comfortable, and efficient operation the noise level of any mechanized vehicle should be kept to a minimum. This may be achieved by the following methods advocated by several sources.

(a) Isolating by the use of soft mounts under the engine and exhaust system piping.

(b) Diminishing by a coating of vibration damping compound on metal panels which reduced noise to a lower and less irritating frequency.

(c) A silencer on the manifold intake.

(d) A muffler which reduces the velocity of expelled exhaust.

Since noise and vibration frequently have common origins, the improvements mentioned above for the reduction of noise should help to decrease the effects of vehicle vibration. The skillful use of absorbents and shock mountings can provide extensive protection.

Although the modern automobile on a good road is an excellent example of a vehicle that is relatively free from vibration, the situation in trucks and some other types of vehicles is quite different. That vibration in such forms of transport affects the performance of the operator can be gaged from the complaints of headaches, fatigue, and eye strain from those subjected to it. See Fig. 4.

C. *Suggested Limits for Variables Influencing Comfort* – A summary of the various stimuli influencing passenger comfort in terms of desirable and maximum values for trucks and buses is presented in Table 3. The highest degree of comfort would be achieved by conforming to the comments listed in the "desirable" column. Discomfort might be expected to arise if the maximum values given in the last column are exceeded. A schematic interpretation of the various stimuli encountered in driving is given in Fig. 5 in terms of limits for comfort, discomfort, and unbearable zones.

The values indicated in the table and figure cannot always be considered as rigid standards because of the interdependence of one variable with others. The difficulty of giving a single numerical value as a standard is exemplified in the limits for carbon monoxide. The generally accepted standard is 0.005% carbon monoxide in circulating air. This figure may be satisfactory for a person sitting

[7] "Tentative Criteria for Noise Control Design," by W. Rosenblith, L. L. Beranek, R. H. Bolt, R. B. Newman, J. J. Baruch, and S. Labate. Presented at the 43rd Meeting of the Acoustical Society of America, New York City, May, 1952.

Table 4 – Characteristics of Brake-Pedal Design and Location in 12 Different 1950-1951 Truck Models

	100	110	120	130	140	160	180	200	220	240	260	280
Type of Pedal (Pedal P or Treadle T)	P	T	T	T	P	P	T	P	T	T	P	T
Pedal Width, in.	4⅜	3	2½	2½	3[b]	3	2½	3¾	3	2¾	3⅝[b]	3
Pedal Length, in.	2⅝	10⅝	10 1/16	10	3	3	10	2¼	10	10¼	3⅝	12
Acute Angle Formed by Pedal Surface and Horizontal, deg	0	65	67	54	—	80	61	86	72	75	67	77
Surface Characteristics.[a]	Rib. Mt.	Wa.Mt.	Rib. Mt.	Rib. Mt.	Rib. Rub.	Rib. Mt.	Sm. Mt.	Rib. Mt.	Rib. Rub.	Rib. Mt.	Rib. Rub.	Rib. Mt.
Height of Pedal Above Floor Horizontal, in.	8⅜	—	—	—	10½	10¼	—	8¼	—	—	9	—

[a] Rib.=Ribbed; Wa.=Waffle; Mt.=Metal; Sm.=Smooth; Rub.=Rubber.
[b] "140" brake pedal is round; "260" brake pedal is round.

Table 5 – Amount of Seat Adjustability in 12 Different 1950-1951 Truck Models

	100	110	120	130	140	160	180	200	220	240	260	280
Vertical, in.	3	0	0	0	0	⅝	0	0	0	1½	0	0
Horizontal, in.	0	4	3½	4	3	4⅛	4½	4	4	5	2	0
Angulation of Front of seat, in.	0	0	0	0	0	0	0	0	0	1½	0	0
Angulation of Rear of Seat, in.	0	0	0	0	0	0	0	0	0	1½	0	0
Angulation of Top of Seat Back, in.	3	0	0	0	½	0	0	0	0	0	0	0
Angulation of Bottom of Seat Back, in.	3	0	0	0	½[a]	0	0	0	0	¾	0	0

[a] Top and bottom of seat back move together.

quietly at sea level, but it should be reduced if the amount of exercise, degree of ventilation, or the length of time increases. The same complexity characterizes almost every other function that has been discussed.

Summary Evaluation of 12 Vehicles

After a procedure for evaluating vehicles in terms of the factors outlined here was established and the body dimensions of the drivers ascertained, it was possible to complete another phase of the investigation which included the evaluation of twelve trucks and six buses with respect to the items mentioned above. All of them were 1950 models, and they represented various types such as light and heavy tractors, straight trucks with cab-over-engine design, both gas and diesel, and sleeper cabs. All of the buses studied had rear engines, some of which were diesel. After these findings have been surveyed by the manufacturing companies, specifications for redesign in subsequent models will be formulated. Obviously, many features of the working area and instrumentation of trucks and buses should be standardized if drivers are to be transferred from one model to another without becoming involved in errors. Table 4 is presented as an example to show the wide range of differences in the brake design of twelve trucks, while Table 5 shows the lack of adequate seat adjustability in the majority of the trucks evaluated. Only three have any vertical adjustability, while two are lacking in horizontal adjustability. Those two tables have been taken from a recently published report, which applies in detail the methods outlined here.[8]

In general the findings of our evaluations were borne out by 250 interviewed commercial drivers. Although the drivers liked and were well adapted to their vehicles, they did make some useful and interesting suggestions. The important ones are listed below.

1. *Vision* – Comments on defective vision were three times as frequent with bus drivers as with the truck drivers. The bus drivers were concerned with blind spots due to large rear-view mirrors,

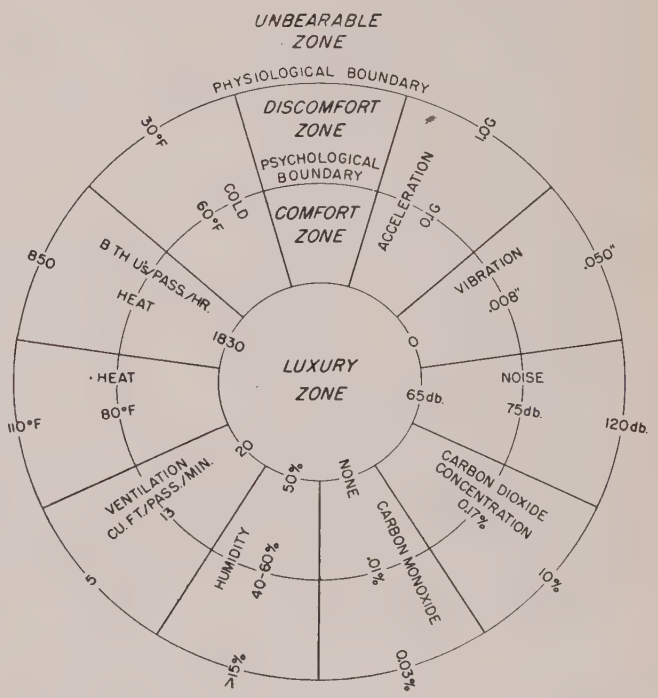


Fig. 5 – Limits for physical variables influencing drivers and passengers. These values must be considered as approximations since each variable is interdependent on one or more additional factors

[8] See "Human Factors in the Design of Highway Transport Equipment: A Summary Report of Vehicle Evaluation," by R. A. McFarland, J. W. Dunlap, W. A. Hall, and A. L. Moseley. Published by Harvard School of Public Health, Boston, 1953.

and truck drivers referred to blind spots due to the door posts in front. Defective windshield cleaning was referred to frequently, and the value of curved and colored windshields was questioned.

2. *Pedals* – Pedals were found to be defective in surface characteristics, size, angulation, and pressure requirements for operation. Some difficulties in the design of brake and accelerator pedals were found to be related to efficiency in braking.

3. *Hand Brakes* – Bus drivers preferred a pull-back type of brake located on the left side of the cab. Truck drivers tended to prefer a pull-up type of hand brake located on the right. Significant comments were made on the inadequate design of hand brakes resulting in low "holding" ability and introducing the possibility of error in dual-range vehicles.

4. *Seats* – Truck – As to ranges of adjustability for angulation – vertical and horizontal adjustments – 69% of the truck drivers desired better adjustment.

Bus – More angulation was desired by 27% of the bus drivers. Bus drivers also desired more shock absorption and better ride qualities in the seat, with more adequate curvature in the seat back.

5. *Steering Wheel* – Comments on steering wheel included:

(a) Requests to incline the wheel to promote efficient arm motion while seated.

(b) Difficulty in steering at low speeds and during blowout maneuvers.

(c) Other data indicated poor road-shock characteristics and a tendency for brief loss of control when the vehicle encountered "pot holes" or stones in the roadway.

6. *Instruments* – Comments by a small number of drivers indicate that instruments are not legible, are poorly illuminated, improperly located with respect to visibility needs, and are improperly grouped. More adequate devices for warning of system failure are needed. Indicator needles are not uniform in the direction of travel so that errors in reading instruments occur.

7. *Switches* – Switches are felt to be the most dependable mechanical units in the vehicle, but their location, especially those needed while in motion, needs to be reconsidered. Switches should be standardized as to location and should be accessible without obstruction.

8. *Brakes* – Brake lag shows considerable variability so that the driver can never predict his total stopping distance. Both truck and bus drivers would like to have this lag reduced to a constant minimum. The mechanism for adjusting brakes is not easily accessible on the rear wheels of some buses, and there is a tendency for mechanics in secondary terminals to evade adjusting them.

9. *Temperature Controls and Ventilation* – Many comments indicate that winter temperature controls need improvement, both in trucks and buses. In general, the buses are heated more adequately, although the maintenance of thermostats is unsatisfactory. In trucks, the firewall conducts a great amount of heat during the summer months, which suggests a change in the design of the forward air vents. Bus windows which open to the front of the vehicle develop an annoying noise characteristic at speeds above 43 mph and have to be closed to control the noise. Some air vents are permanently open and continual intake of cold air in the winter causes discomfort. The location of the air intake in some vehicles is such that it permits intake of exhaust from vehicles ahead.

10. *Passenger Entry* – Icing conditions on the bottom step suggest the need for heating the step. Hand rails are not designed with passenger needs in mind. The outside step is too high from the ground to meet the needs of very heavy or elderly passengers.

Summary

The high incidence of temporary and fatal injuries resulting from highway accidents today constitutes an important problem for the truck and bus industries. Since it is known that human factors play an important role in the causation of accidents, it logically follows that control measures should be broadened to include contributions from the human engineering aspects of the biological and psychological sciences.

With this goal in view the Harvard School of Public Health has had an extensive research program in the field of highway safety for the past four years, in which the promising areas of investigation are as follows: (1) job and activity analysis including the study of near accidents, (2) health maintenance examination, (3) detection of the accident repeater, (4) personal adjustments, (5) morale, (6) supervision and leadership, (7) human body size and capabilities with reference to the design and layout of all controls, instruments, seats, and areas of vision within the cab, (8) the effect of physical variables on the driver such as temperature, humidity and ventilation, noise and vibration, and finally (9) safety features to protect the operator if an accident occurs.

It is believed that such an approach, concerned with the human factors of design and operation can help to reduce the high frequency of accidents, lower training and operating costs, and prevent the extensive redesign of equipment after it is put in use.

Build Safety Into Design of Car

– Arthur W. Stevens
Automobile Safety Association

THIS paper is especially valuable because it emphasizes the necessity of designing commercial equipment to fit the operator.

This same principle applies even more to the design of noncommercial automobiles, the drivers of which are not subject to continuing screening as to qualifications ordinarily applied to commercial drivers.

Auto makers must recognize that their product is not going to be operated by a driver who is ideal physically and mentally, perfect as to eyesight and reactions, always sound as to judgment. Auto makers must not count even on the car's being operated by the mythical "average driver." They must plan and design passenger autos for all kinds of drivers, young and old, tall and short, alert and dull, fresh and fatigued.

Safety must be built into the design of the car.

Today's car is so built that the operator is unable to see to drive safely. Seated back toward the rear of his car and down behind a high, view-obstructing hood, wide corner posts, and bulging fenders, he is robbed by the sheet metal of his own car of his view of the road close up ahead and at the sides. He must guess at clearances and intervals. He overruns stop lines, straddles lane lines, and may even start up his car and run over his own or his neighbor's 5-year-old child playing unseen and unseeable in front of the radiator.

Why does the industry continue to load down the front end of every car with the heavy engine? With a front-end-heavy car, when brakes are applied to decelerate the car sharply, the heavy front end is depressed and the rear end lifts. The industry advertises its 60-40 brakes as effective for "straight-line stops." If you try to slow down quickly and at the same time try to avoid hitting someone or something or even to negotiate a curve, you take the chance of your rear tires completely losing traction, with a resultant skid and overturn.

The glaring headlights, placed so low that they must shine up into and blind the eyes of approaching motorists or pedestrians; the inverted chimney exhaust which vents deadly carbon monoxide at ankle height where it's most liable to affect pedestrians or be drawn into the car immediately behind—these are other dangerous features which must be eliminated if automobile accidents are to be brought under control.

ORAL DISCUSSION

Reported by Glenn W. Johnson

Bowman Dairy Co.

L. C. Kibbee, American Trucking Associations, Inc.: Isn't it better to segregate instruments as to their importance: (1) those you have to read while driving and (2) those that can be read after the truck stops?

Dr. McFarland: I agree, and I think a study should be made to decide which are the critical instruments as there seems to be a variance of opinion. It would be possible to determine the most important instruments by recording the eye movements of a series of professional drivers while at work. Such a procedure has been used with marked success in determining the relative importance of instruments in aircraft panels. The recent report of the SAE Committee on Cockpit Standardization (S-7) should be consulted.

Robert Cass, White Motor Co.: Do you feel that you have gone far enough with your study on cab design to recommend space in cab, location of steering wheel, location of instruments, and other similar items?

Dr. McFarland: There are recommendations in our manual[5] relating to clearances, seating, and certain other items. Objective data on the location of the steering wheel, pedals, and other controls and instruments are being compiled by members of our staff by means of mockups such as that illustrated in Fig. 2.

A. K. Simons, Bostrom Mfg. Co.: What effect do you feel that the role of vibration of the cab has on driver fatigue, accidents, and health?

Dr. McFarland: There is some evidence that excessive amounts of vibration may give rise to lowered operational efficiency and certain clinical conditions, especially of an orthopedic and visual nature. This evidence has been reviewed in the manual mentioned above.

Weather or Knock—

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THE title "Weather or Knock," while somewhat whimsical, was chosen because it portrayed so well the background which led to this paper. The Standard Oil Co. (Ohio) had carried out field work that indicated that the antiknock requirements of vehicles might vary over a wide range as a result of seasonal variations in weather. Ethyl Corp. had conducted a research program designed to study the influence of weather on knock ratings and vehicle requirements in conjunction with its program to set up all-weather rooms for vehicle knock ratings. It was natural, therefore, that the two companies should join forces to explore further and more completely the effects of weather and associated engine variables on the antiknock relationship of fuels and vehicles.

The Standard of Ohio background to this study covered a 3-year period during which the variation in fuel requirements of customer vehicles was studied. As a result of these studies, it was conjectured that such circumstances as the occurrence of warm, dry days combined with the use of high-temperature thermostats and glycol-antifreeze mixtures might cause vehicles to have high antiknock requirements. Conversely, cool, damp weather, which occurred concurrently with the use of water as a coolant and with low-temperature thermostats, might result in low requirements. The test data available from the Ethyl research program supported this background information, hence a cooperative test program was planned to supplement the available information and to explore all the factors which were believed to be of importance in the seasonal variations.

Study of the effects of weather on knocking is not new. Various investigators[1-5] have reported on the effects of barometric pressure, atmospheric temperature, and humidity on the octane-number requirements of vehicles as measured with various types of reference fuels. However, to the authors' knowledge, this is the most comprehensive study

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[1] See API Proceedings, Vol. 18, Section III, 1937, pp. 137-146: "Effect of Altitude on Antiknock Requirement of Cars," by N. MacCoull, K. L. Hollister, and R. C. Crone.

[2] "Antiknock Quality Requirements of High-Compression-Ratio Passenger-Car Engines," by R. W. Scott, G. S. Tobias, and P. L. Haines. Presented at 114th National Meeting of American Chemical Society, Division of Petroleum Chemistry, St. Louis, Sept. 7, 1948.

[3] See *Journal of Research of National Bureau of Standards*, Vol. 28, June, 1942, pp. 713-734: "Effect of Altitude on Knock Rating in CFR Engines," by D. B. Brooks. (Also in 1946 CRC Handbook.)

[4] See *Journal of Research of National Bureau of Standards*, Vol. 45, August, 1950, pp. 148-153: "Effects of Pressure and Other Variables on Determinations of Octane Number," by W. J. Levedahl.

[5] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 557-570: "Factors Affecting Octane-Number Requirement," by H. J. Gibson.

Road Ratings and Requirements, Year-Round

R. I. Potter and E. H. Scott
Standard Oil Co. (Ohio)

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Paper presented at SAE National Fuels and Lubricants Meeting, Chicago, Nov. 6, 1953.

SEVERAL factors are involved in the answer to the question, "How do atmospheric conditions affect the ability of a fuel to satisfy the antiknock requirement of automotive engines?"

As is well known, an increase in atmospheric temperature increases the octane-number requirement of engines. This paper points out, however, that this causes little change in the road octane-number ratings of commercial fuels.

Increasing the absolute humidity has the opposite effect to increasing the temperature and tends to counteract the undesirable effects of changing temperature throughout the various seasons of the year.

Increasing the barometric pressure or decreasing the wind velocity both increase the tendency of commercial fuels to knock.

Factors indirectly related to weather conditions, such as the coolant or thermostat used in an engine, also affect the knocking tendency of a fuel.

Conditions requiring maximum antiknock quality in a fuel are found when the atmospheric effects of low humidity and high temperature are combined with the engine-cooling effects of high-temperature thermostats and glycol antifreezes. Minimum requirements occur when the weather is cool and moist, water is used as a coolant, and a low-temperature thermostat is used. Actual effects vary widely from car to car but may be as great as 15 octane numbers.

that has been reported and is the first to relate the antiknock performance of commercial and reference fuels over such a wide range of conditions.

This paper presents the combined results of the individual and cooperative studies by the two companies, studies which indicate that weather and knock are closely related indeed. It explores the effects of atmospheric temperature, pressure, and humidity on the knock ratings of commercial fuels

Table 1 – Inspection Data of Test Fuels

Fuel Number	Fuel Description	Gravity	Rvp	Distillation Temperature, F							Tel, ml/gal	Octane No.	
				Ibp	10%	30%	50%	70%	90%	Ep		Research	Motor
Fuels Used for Temperature, Humidity, and Coolant Tests:													
273-51	Commercial	62.9	11.0	83	115	169	219	272	339	414	2.51	92.6	83.9
274-51	Commercial	63.4	12.8	83	111	166	220	275	346	416	....	92.2	83.5
277-51	Commercial	61.9	10.4	91	130	181	228	273	336	394	2.94	92.2	82.5
278-51	Commercial	62.5	11.4	89	123	176	224	271	335	404	2.87	92.2	82.3
281-51	Commercial	57.9	10.6	94	130	189	230	270	331	410	2.81	91.7	85.0
282-51	Commercial	58.9	12.0	89	117	182	227	269	333	402	2.74	92.0	85.2
285-51	Commercial	60.4	10.0	94	129	178	227	279	351	414	2.74	92.6	83.6
286-51	Commercial	61.2	10.9	88	124	174	226	280	350	407	2.70	92.6	83.9
141-52	Commercial	60.2	9.8	94	125	174	221	271	353	418	....	91.5	82.7
271-51	Commercial	63.4	11.0	84	120	174	221	269	335	392	....	87.5	80.8
272-51	Commercial	64.1	13.0	82	110	163	213	263	330	394	1.95	87.7	81.1
275-51	Commercial	60.8	10.2	89	127	173	222	271	335	394	1.92	88.0	80.7
276-51	Commercial	61.5	11.7	89	121	170	223	273	337	394	1.85	88.3	80.8
279-51	Commercial	62.2	10.4	92	131	181	225	271	338	414	2.24	86.8	79.9
280-51	Commercial	63.3	11.9	93	132	181	226	272	337	418	2.19	87.0	80.2
283-51	Commercial	60.8	9.8	96	133	178	225	274	340	398	1.34	88.1	80.5
284-51	Commercial	62.3	11.8	90	119	167	217	269	339	398	1.30	88.1	80.5
140-52	Commercial	62.1	11.8	81	119	165	215	267	343	398	....	86.7	79.5
261-51[a]	Commercial	61.0	8.8	100	126	168	216	262	336	401	0	81.1	74.3
262-51	Commercial	....	....	....	....	....	....	....	....	....	0.19	82.9	75.8
263-51	Commercial	....	....	....	....	....	....	....	....	....	0.44	85.3	77.6
264-51	Commercial	....	....	....	....	....	....	....	....	....	1.02	87.4	78.7
265-51	Commercial	....	....	....	....	....	....	....	....	....	1.82	89.8	80.6
266-51[b]	Commercial	61.0	8.5	94	126	172	217	263	342	404	0.27	87.5	77.9
267-51	Commercial	....	....	....	....	....	....	....	....	....	0.68	89.8	79.9
268-51	Commercial	....	....	....	....	....	....	....	....	....	1.64	92.3	81.6
269-51	Commercial	....	....	....	....	....	....	....	....	....	3.09	94.8	83.9
270-51	Commercial	....	....	....	....	....	....	....	....	....	9.56	98.3	87.2
Fuels Used for Temperature, Humidity, and Wind Velocity Tests:													
427C-50	Catalytically cracked and straightrun	58.8	8.6	91	124	163	220	302	379	425	0.88	90.7	81.1
278-50	RMFD 32-50 high-sensitivity premium	....	....	....	....	....	....	....	....	....	0.61	92.1	81.4
508-50	Catalytically cracked A	58.7	9.4	91	122	161	214	292	379	423	0	92.2	80.0
799-50	65% DIB, 35% n-heptane	....	....	....	....	....	....	....	....	....	0	91.6	80.7
635FF-48	Leaded straightrun	66.2	8.5	94	139	195	239	287	358	406	2.10	91.7	91.7
492N-48	Catalytically cracked B	57.3	9.1	89	120	165	236	309	359	408	0.58	91.4	81.9
Fuels Used for Barometric Pressure Tests:													
A	Commercial premium	....	....	....	....	....	....	....	....	....	1.32	88.2	80.3
B	Catalytically cracked	....	....	....	....	....	....	....	....	....	0	91.6	80.7
C	Catalytically cracked	....	....	....	....	....	....	....	....	....	0	91.1	82.4
D	63% RMFD 37-50, 37% RMFD 38-50	....	....	....	....	....	....	....	....	....	1.89	88.4	81.5
E	Commercial premium	....	....	....	....	....	....	....	....	....	1.34	91.0	81.5
F	Commercial premium	....	....	....	....	....	....	....	....	....	2.98	93.1	83.5
G	Commercial premium	....	....	....	....	....	....	....	....	....	2.94	91.2	83.3
H	Commercial premium	....	....	....	....	....	....	....	....	....	2.82	90.5	83.1

[a] No. 1 base gasoline stock for various tel concentrations.
[b] No. 2 base gasoline stock for various tel concentrations.

and, more important, on their ability to satisfy the vehicle. The possible effects of wind velocity are shown, as are the effects of the variables in engine cooling: the thermostat setting and the type of coolant used.

Factors Affecting Seasonal Antiknock Variations

The factors investigated were:

1. Climatic conditions:
   a. Ambient air temperature.
   b. Humidity.
   c. Barometric pressure.
   d. Wind velocity.
2. Engine cooling:
   a. Thermostat operating temperature.
   b. Coolant type.
3. Fuel properties:
   a. Reid vapor pressure.
   b. Composition.

Three of the principal headings shown under "Climatic conditions," air temperature, pressure, and humidity, seemed of obvious primary importance, while wind velocity, which has a bearing on engine temperatures, was believed to be of secondary importance. Under "Engine cooling" came the thermostat setting; many people use high-temperature thermostats to improve car heater performance or to raise the general level of engine temperatures. Coolant type was another important variable in this category. The use of permanent, glycol-type antifreeze solutions is common, particularly in the more expensive, high-requirement cars. The heat-transfer characteristics of these coolants are known to raise metal-surface temperatures relative to those obtained with water. The

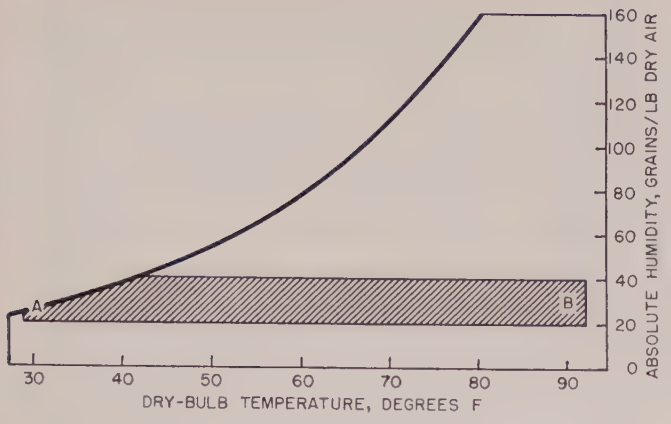


Fig. 1 – Portion of psychrometric chart explored for temperature effect

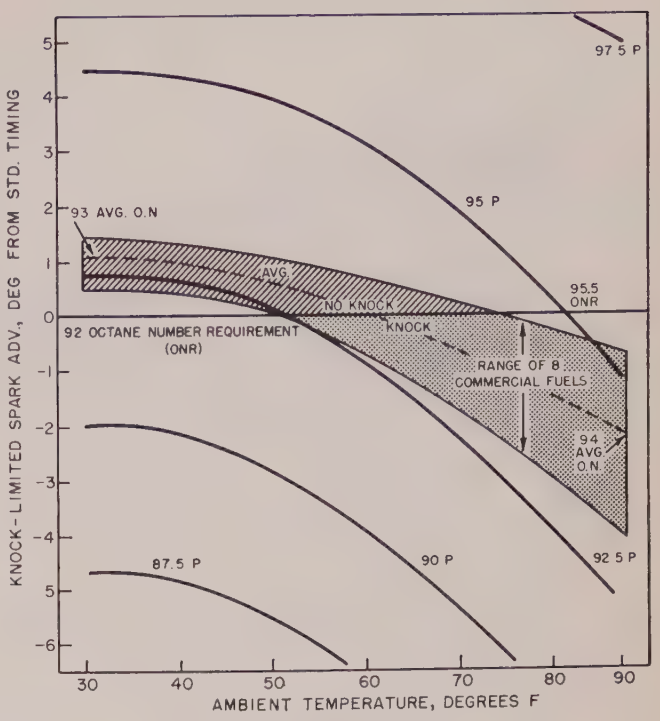


Fig. 2 – Effect of change in temperature on fuel – knock humidity 20 grains per lb, car$_2$, average of two coolant and thermostat combinations

combination of a high-temperature thermostat and a glycol-type coolant would be expected to result in high engine temperatures.

None of the foregoing information would be of much practical value, however, unless it was expressed in terms of commercial gasolines as well as the usual primary reference fuels. Therefore, a wide variety of commercial gasolines, both premium and regular, and a variety of special fuels were used in the tests.

Test Equipment and Procedure

Test cars of nine different makes and models were used. They were: 1949 Chevrolet, 1950 Ford V-8, 1950 Cadillac, 1949 Cadillac, 1949 Oldsmobile 88, 1949 Oldsmobile 76, 1950 Buick 40 with manual shift, 1950 Buick 40 with Dynaflow, and 1948 Pontiac 8 with Hydra-matic.

Three of these were selected because field experience indicated that they were particularly sensitive to seasonal changes. The others were vehicles considered to be of general interest.

The effects of temperature, absolute humidity, wind velocity, thermostat, and coolant were studied in a chassis dynamometer room where atmospheric variables could be controlled independently.

The observations on the effect of barometric pressure were made in the vicinity of San Bernardino, Calif., at altitudes from sea level to 6000 ft (30 to 24 in. Hg barometric pressure) with conditions selected to give essentially constant temperature and humidity.

All of the fuel ratings and engine requirements were based on data obtained by the Modified Uniontown Technique (CRC Designation F-28). In this technique a fuel is rated in terms of the degrees of spark advance that the fuel will tolerate before it gives trace detonation. The greater the number of degrees of spark advance that a fuel will tolerate, the higher its antiknock value. The degrees of spark advance for trace detonation have been termed "knock-limited spark advance" throughout this paper. It should be pointed out that when fuel ratings have changed as a result of engine or atmospheric variables, it is possible that some of the difference was a result of the change in ignition timing, since this is known to influence the severity of some engines.

Test Fuels

A variety of test fuels, both commercial and special, was used. They covered a wide range of refining processes, tetraethyl lead contents and volatility characteristics. Pertinent inspection data are shown in Table 1.

Effect of Ambient Air Temperature

The effect of changes in ambient air temperature on the knocking of fuels was studied in several makes of cars. The temperatures used ranged from 30 to 90 F with an absolute humidity of 20 to 40 grains of moisture per lb of dry air, as shown in the psychrometric chart, Fig. 1. The effect of changing ambient air temperature on the knocking tendency of fuels in car E_2 is shown in Fig. 2. The variation in the ability of fuels to satisfy the antiknock requirement of an engine is illustrated

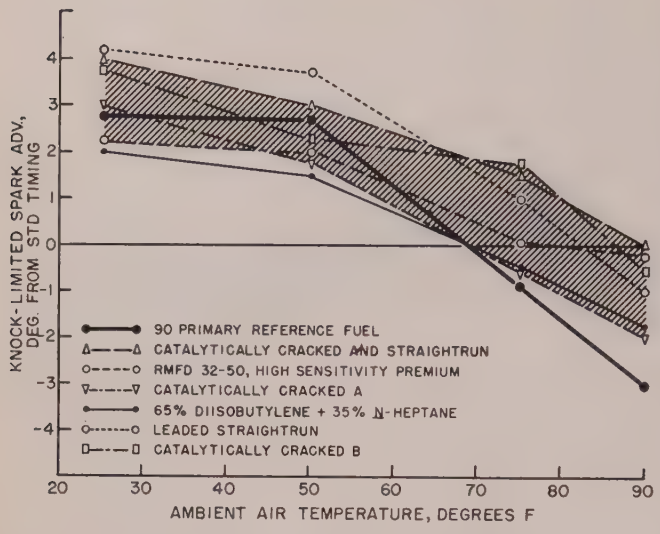

KNOCK-LIMITED SPARK ADV., DEG. FROM STD TIMING
90 PRIMARY REFERENCE FUEL
CATALYTICALLY CRACKED AND STRAIGHTRUN
RMFD 32-50, HIGH SENSITIVITY PREMIUM
CATALYTICALLY CRACKED A
65% DIISOBUTYLENE + 35% N-HEPTANE
LEADED STRAIGHTRUN
CATALYTICALLY CRACKED B
AMBIENT AIR TEMPERATURE, DEGREES F

Fig. 3 – Effect of change in temperature on fuel knock – humidity 30 grains per lb, glycol-water coolant, car E_1, average of two ratings

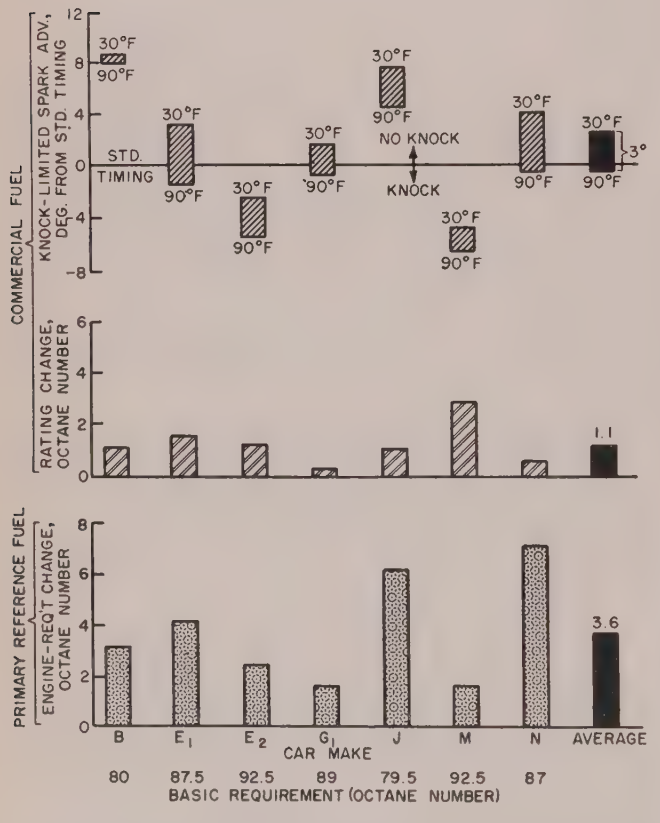


Fig. 4 – Effect of temperature change of 60 deg on fuel knock – temperature changed from 30 F to 90 F, humidity 20-40 grains per lb

by showing the change in the knock-limited spark advance of fuels as the temperature is changed.

The standard ignition timing of the engine is indicated by the zero line on the curve. Conditions which allow the fuels to have knock-limited spark advance above this line are ones that will permit knock-free operation at standard ignition timing. Conditions which cause the knock-limited spark advance to fall below standard timing, however, are ones which cause the fuel to be unsatisfactory in antiknock quality for this engine. Determination of these latter conditions was a primary objective of this investigation. All eight of the commercial fuels satisfied the engine of car E_2 at low atmospheric temperatures, but when the temperature was increased, at constant low humidity, the knock-limited spark advance became less than standard timing and, therefore, caused the fuels to become too low in antiknock value to satisfy the engine requirement. It should be observed that the knock-limited spark advance changed more rapidly with temperature for primary reference fuels than for the commercial fuels. As a result, the octane-number ratings of the commercial fuels *increased* even though the ability of the fuels to satisfy the engine *decreased.*

The average vehicle-octane-number rating of the eight fuels was 93 at 30 F and 94 at 90 F, as indicated by their position relative to the primary reference fuels. The position of the curves for the primary reference fuels in relation to the horizontal standard timing line shows that the octane-number requirement of the engine changed from 92 to 95.5 as a result of the 60 F temperature change. When the 1.0 octane-number increase in rating of the fuels is subtracted from the 3.5 octane-number increase in engine requirement, the net *decrease* in the ability of the fuels to satisfy the engine was 2.5 octane numbers. This illustrates the importance of relating fuel-octane-number ratings to engine requirement *at the same temperature* if the ability of a fuel to satisfy the engine in which it is being used is to be completely evaluated.

Similar data for a group of special fuels run in another car of the same make are shown in Fig. 3. Results for several makes of cars are summarized by the bar charts in Fig. 4. The portion of this figure for the knock-limited spark advance shows that cars B and J were satisfied by commercial fuel throughout the 30 to 90 F temperature range since the knock-limited spark advance was above the standard timing line. Cars E_1, G_1, and N gave knock-free operation at 30 F, but were knocking on the fuel at 90 F. Cars E_2 and M had high initial octane-number requirements and were in the knocking range with the commercial fuel at all temperatures from 30 to 90 F.

The fuel octane-number rating was higher at 90 F than at 30 F, while the engine octane-number requirement increased considerably with the 60-deg increase in ambient air temperature. The

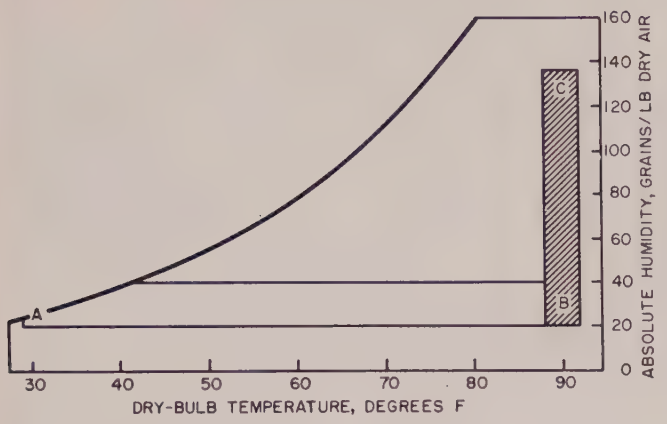


Fig. 5 – Portion of psychrometric chart explored for humidity effect

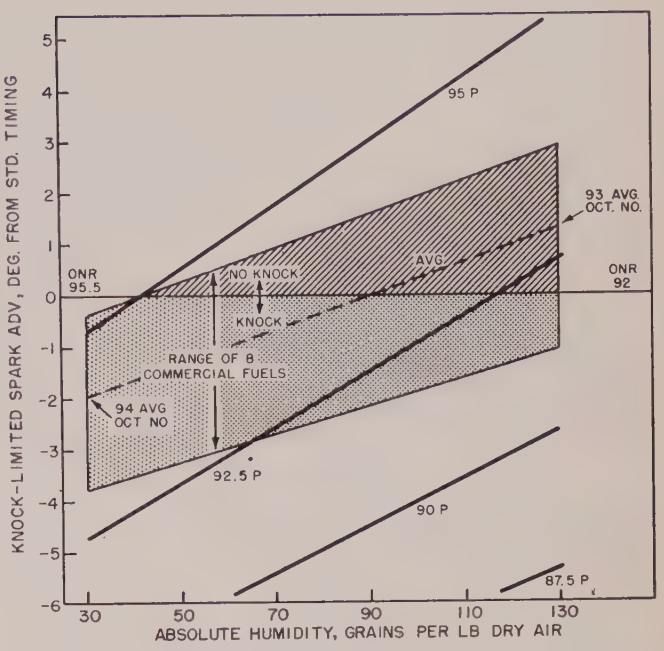


Fig. 6 – Effect of change in humidity on fuel knock – constant temperature (90 F ± 2 deg), car E_2, average of two coolant and thermostat combinations

average effect was to decrease the knock-limited spark advance 3 deg, increase the rating of the fuels 1.1 octane numbers, and increase the octane-number requirement of the engines 3.6 octane numbers. The difference between 3.6 and 1.1 is an average 2.5 octane-number *loss* in the antiknock value of the fuel relative to engine requirement, thus showing that even though the fuels were rated *higher* in terms of primary reference fuels in all cars, the interpretation of the engine was that they became *poorer* as the ambient air temperature changed from 30 to 90 F.

Effect of Humidity

The next phase of this investigation was a study of the effect of absolute humidity on fuel knocking. The humidity range from 30 to 130 grains moisture per lb of dry air, shown graphically on the psychrometric chart in Fig. 5, was investigated. Commercial and reference fuels were run in several makes of cars with the ambient temperature held constant at 90 F ±2 deg, while the absolute humidity was varied.

The effect of changing the absolute humidity on the knocking tendency of fuels in car E_2 is shown in Fig. 6. Increasing the humidity had an effect opposite to that of increasing the temperature. Knock-limited spark advance was increased as the humidity was increased. The rate of change in the knock-limited spark advance was again less for commercial fuels than for primary reference fuels. The average octane-number rating of the commercial fuels changed from 94 at 30 grains to 93 at 130 grains for a *decrease* of 1.0 octane number for the 100 grains increase in absolute humidity. A simultaneous decrease of 3.5 octane numbers, from 95.5 to 92, in engine requirement was obtained, as shown by the position of the lines for the primary

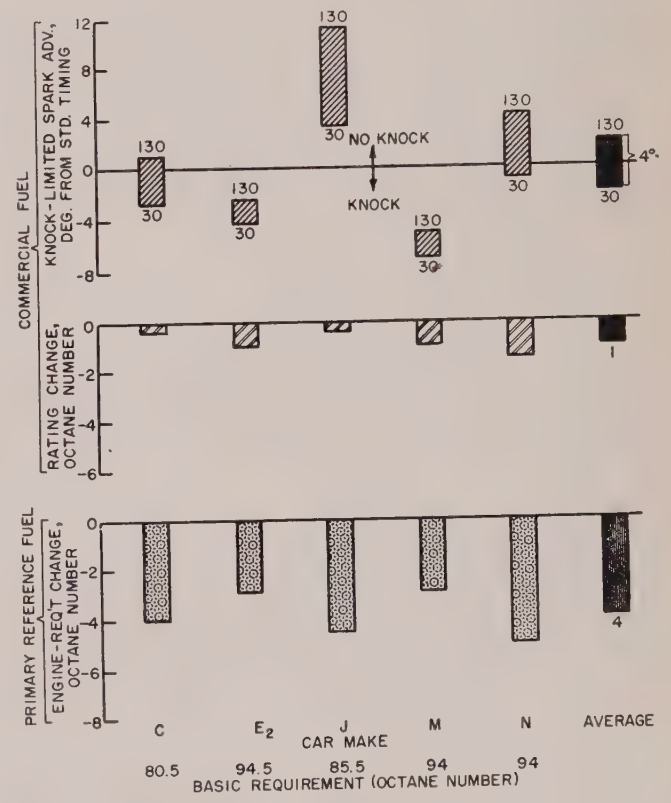


Fig. 7 – Effect of humidity change of 100 grains on fuel knock – humidity changed from 30 to 130 grains per lb, constant temperature 90 F ± 2 deg)

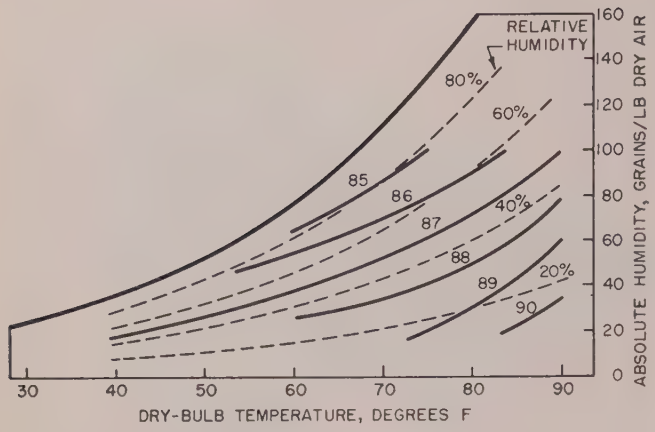


Fig. 8 – Octane-number requirement within psychrometric area – commercial fuels, car C

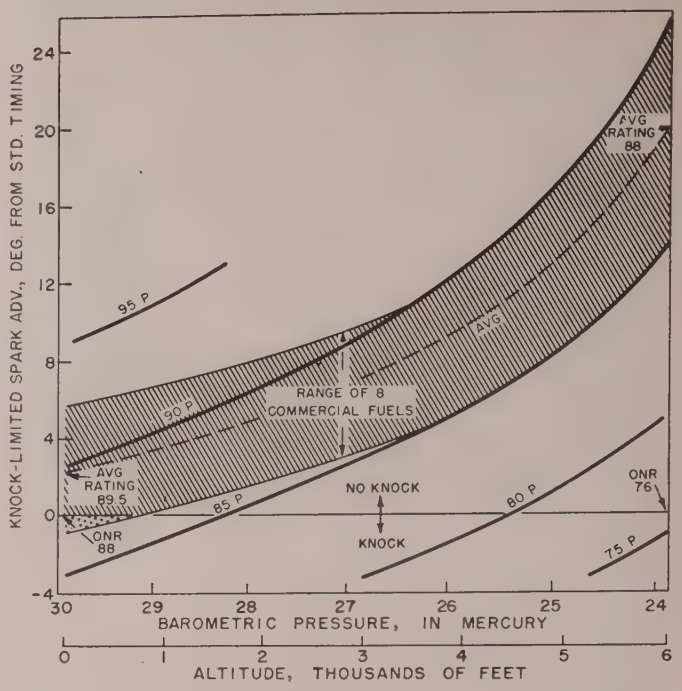


Fig. 10 – Effect of change in barometric pressure on fuel knock – temperature 70 F, humidity 30 grains per lb, car E

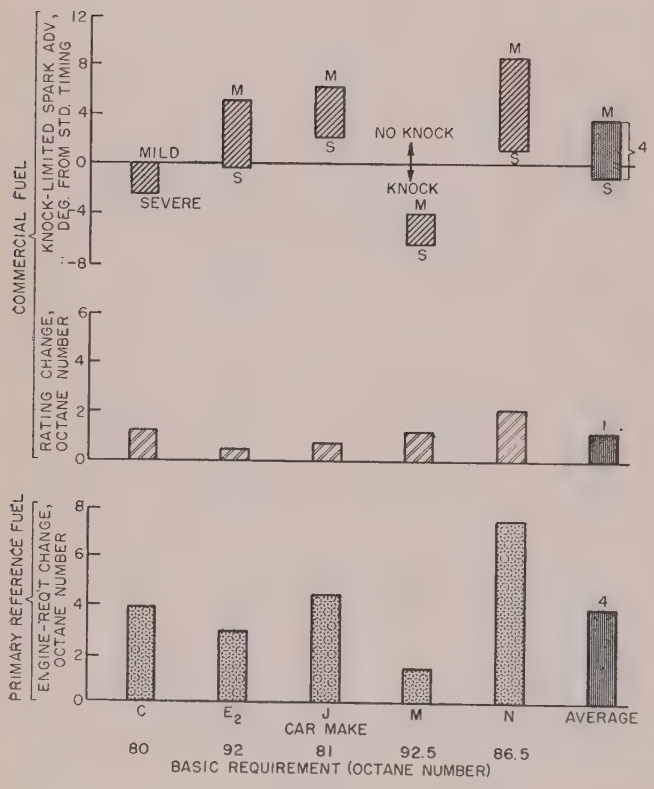


Fig. 9 – Combined effects of change in temperature and humidity on fuel knock – change from mild conditions (50 F, high humidity) to severe conditions (90 F, low humidity)

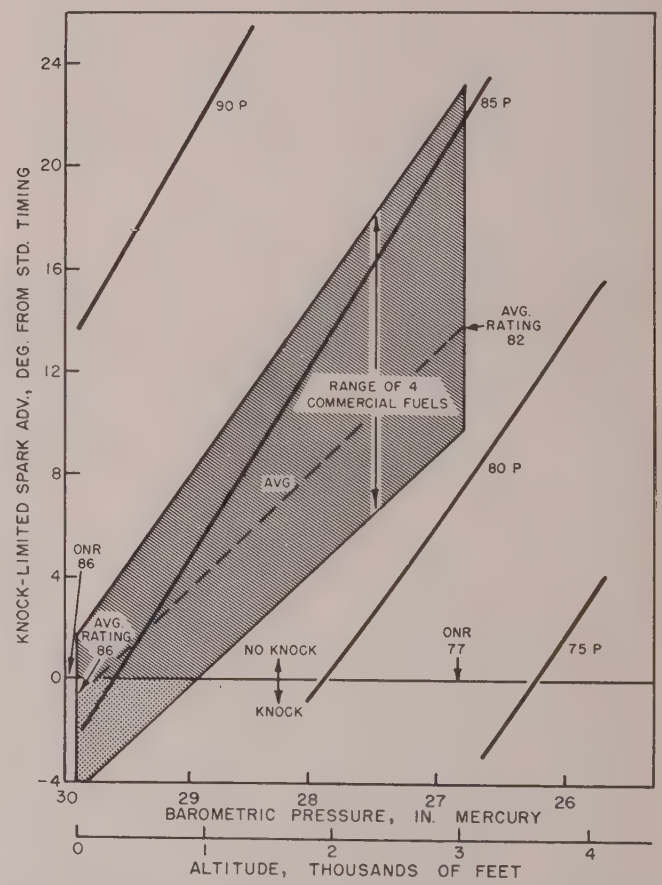


Fig. 11 – Effect of change in barometric pressure on fuel knock – temperature 70 F, humidity 30 grains per lb, car M_1

reference fuel relative to the standard timing line. The 2.5-octane-number difference between the 3.5- and 1.0-octane-number changes is the improvement in the ability of the average fuel to satisfy the engine.

The effect of an 100-grain increase in absolute humidity for several makes of cars is shown in Fig. 7. The knock-limited spark advance for a 90-octane-number commercial fuel increased from 2 deg for cars E_2 and M to approximately 8 deg for car J. Octane-number ratings decreased 0.5 to 1.5 octane numbers, and engine requirement decreased 3 to 5 octane numbers. The average effect was to increase the knock-limited spark advance 4 deg, decrease the relative ratings of the commercial fuels 1.0 octane number, and decrease the engine requirement 4.0 octane numbers. This gave an average gain of 3.0 octane numbers in the anti-knock effectiveness of the commercial fuel for the 100-grain increase in absolute humidity.

Combined Effect of Temperature and Humidity

To obtain an overall picture of how the octane-number requirement of one engine would vary throughout the psychrometric area, this requirement was measured at many combinations of temperature and humidity other than those used for survey purposes. The results of these measurements with commercial fuels are shown as lines of constant Research octane-number requirement in Fig. 8. It is significant to note that these lines are similar in shape to the lines for constant relative humidity on the psychrometric chart. This was also found to be true with primary reference fuels. At approximately constant *relative* humidity, ratings of fuels and the octane-number requirements of cars would tend to remain nearly constant over a wide temperature range. This is particularly true in the range of temperature and humidity most frequently experienced. However, it can be seen that the requirement of this car was changed by as much as five octane numbers by variations in temperature and humidity.

Fig. 9 shows the combined effects of change of ambient temperature and humidity for all the cars tested. The top graph shows that the change in knock-limited spark advance with a commercial fuel varied from a minimum of 2.5 to a maximum of 7 deg. In all, the five cars changed an average of 4.5 deg over the range of weather covered on the psychrometric chart. Similarly, for these same conditions the ratings of the commercial fuel changed a maximum of 2.0 octane numbers, with an average of 1.0 octane number for the five cars. The third graph shows that the change in engine requirement with primary fuels varied from 1.5 to 7.5, for an average 4.0 octane numbers. The importance of this change due to the combined effects of temperature and humidity is emphasized when it is realized that this variation is, in many cases, roughly equivalent to the difference in octane number between regular- and premium-grade gasolines. It demonstrates also that many fuels normally satisfactory for engine operation may become inadequate with the rise in engine octane-number requirement encountered under adverse temperature and humidity conditions.

From the standpoint of road-test techniques, the data indicate that compensation or correction for weather conditions should be made on the basis of frequent weather observations.

Effect of Barometric Pressure

The effect of changing barometric pressure on fuel knock and engine requirement is shown in Fig. 10 for car E. Fuel ratings made at sea level and at altitudes up to approximately 6000 ft permitted evaluation of the effect of a change in barometric pressure of 6 in. of Hg. Temperature and humidity were kept approximately constant by selection of conditions at the time of test. Going from 30 in. Hg to 27 in. Hg barometric pressure increased the average knock-limited spark advance for commercial fuels by 4 deg, while a further decrease to 24 in. increased this spark advance another 13 deg. The average rating of the commercial fuels decreased from 89.5 to 88, but the engine requirement decreased from 88 to 76 octane numbers for the change from 30 to 24 in. Hg.

Similar data are shown in Fig. 11 for car M_1. This car decreased in requirement more rapidly than car E, hence data were obtained only to an altitude of 3000 ft or a barometric pressure of 27 in. Hg. Going from 30 to 27 in. Hg increased the average knock-limited spark advance for commercial fuels by 13 deg; the average octane-number ratings were decreased by 4 octane numbers; and engine requirement decreased by 9 octane numbers.

The results for these two cars are summarized in Fig. 12, which indicates that in the range of 30 to 27 in. Hg a change in barometric pressure of 3.0 in. allowed the knock-limited spark advance for commercial fuel to be increased an average of 8.5 deg, whereas the ratings were decreased an average of 2.5 octane numbers. The engine requirements decreased an average of 7.0 octane numbers.

Effect of Wind Velocity on Fuel Knock

The effect of changing wind velocity from a 10-mph tail wind to a 10-mph head wind, designated by the T's and H's on the bar chart in Fig. 13, was to increase the knock-limited spark advance of commercial fuels from 0 to 3.5 deg, to give no significant change in fuel octane-number ratings, and to decrease engine requirement 1 to 4 octane numbers in the three cars B, E, and G_1. The small change in the octane-number rating of the fuels should not be interpreted as indicating that wind direction need not be considered during fuel ratings, however. In obtaining the results shown, the reference fuels were run under conditions identical

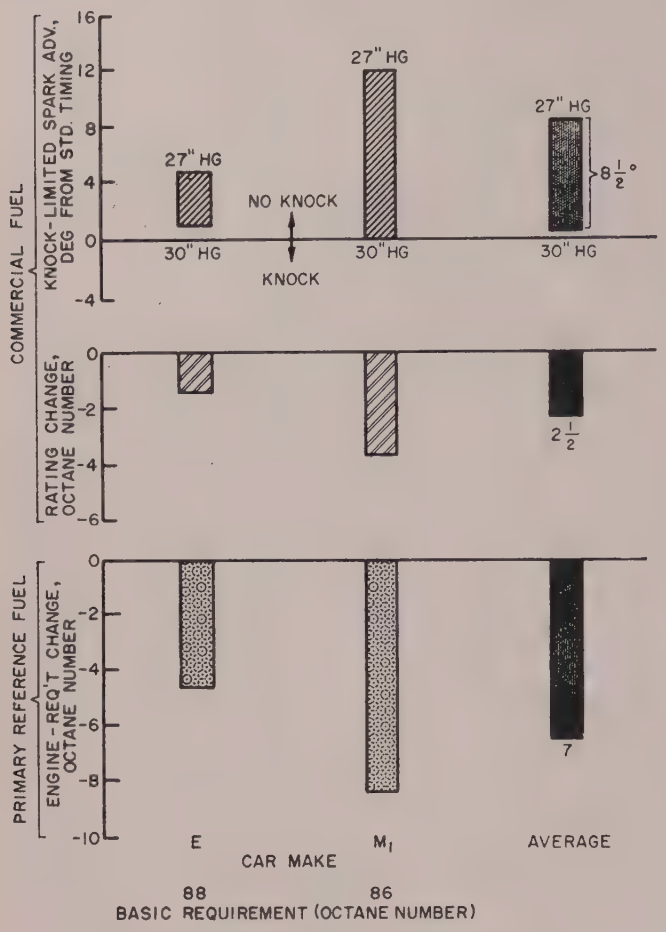


Fig. 12 – Effect of change in barometric pressure on fuel knock – change from sea level (30 in. Hg) to 3000-ft altitude (27 in. Hg), temperature 70 F, humidity 30 grains per lb

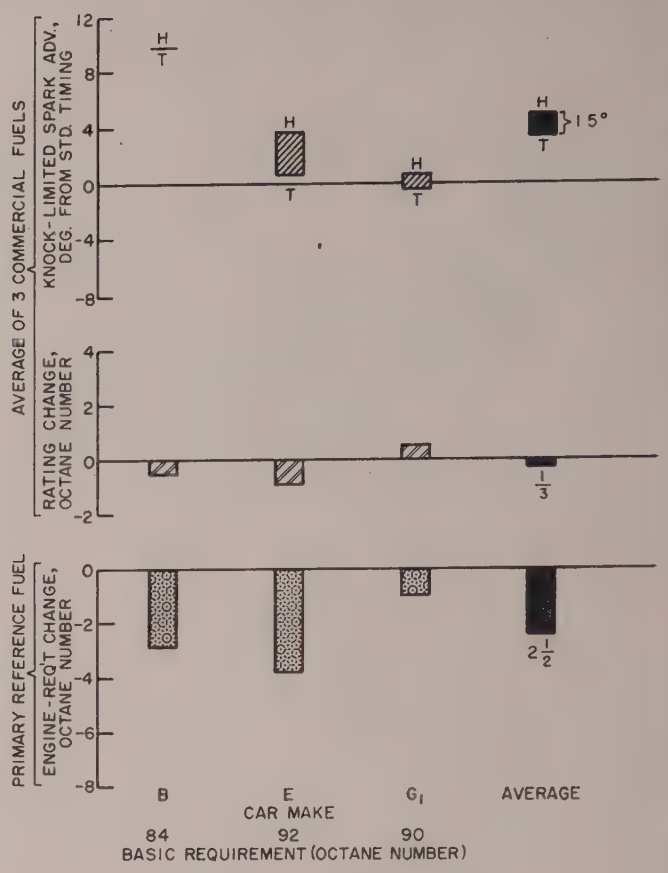


Fig. 13 – Effect of change in wind velocity on fuel knock – temperature 75 F, humidity 30 grains per lb, change from 10-mph tail wind (T) to 10-mph head wind (H)

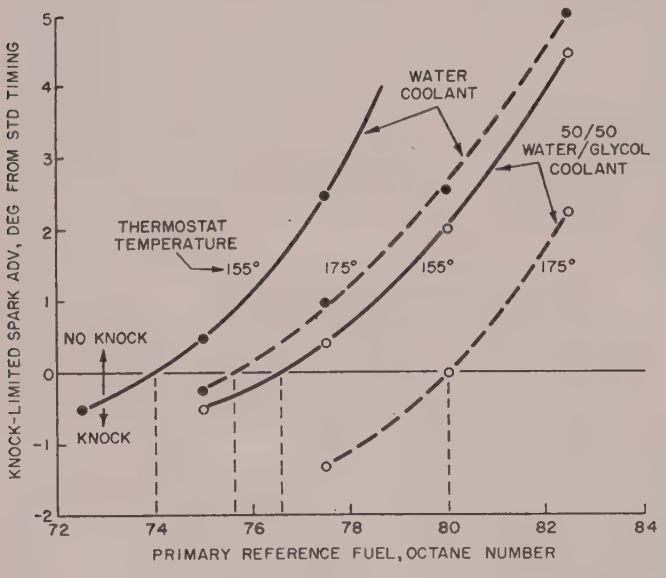


Fig. 14 – Effect of changes in thermostat and coolant on fuel knock – temperature 50 F, humidity 33 grains per lb, car J

to those used for the commercial fuels. Since there was a definite change in engine requirement with the change of wind velocity, it can readily be seen that an error in fuel octane-number ratings would exist if, for example, reference fuels were rated while driving in a head wind and test fuels were rated while driving in a tail wind. Furthermore, these results indicate that care must be used by road-rating crews in spotting and compensating for changes in wind direction which occur during the work day.

Engine Cooling System Variables

The effects of variables in the engine cooling system on vehicle octane-number requirement and fuel rating under various climatic conditions were studied by making tests with:

1. Low- and high-temperature-range thermostats.
2. 100% water and 50/50 water-glycol mixture coolants.

These conditions represent possible extremes of thermostat and coolant combinations. However,

one of the major questions that arose in this program was the possible effect of season-to-season carryover of these coolant-thermostat combinations.

The data to show the effect of thermostat operating temperature, the effect of coolant composition, and the effect of various combinations of thermostat and coolant combinations on vehicle octane requirement at various ignition spark settings were obtained at an ambient temperature of 50 F and a humidity of 33 grains per lb. This combination of temperature and humidity typifies a clear, crisp spring or fall morning.

In Fig. 14 is shown the effect of primary reference fuel octane number on knock-limited spark advance in car J with: (a) two different thermostat settings, and (b) two different coolants, water and a 50/50 water-glycol solution. This figure shows that as the operating temperature of the thermostat was increased, the octane-number requirement was increased, for any given spark setting. At the factory standard spark setting this change in octane requirement was 1.5 octane numbers with water and 3.5 octane numbers with glycol-water. Similarly, the effect of change from water to glycol-water coolant with the low-temperature thermostat was 2.5 octane numbers, while with the high-temperature thermostat the change was 4.5 octane numbers. The overall change was 74 to 80, a range of 6 octane numbers.

In Fig. 15 similar data are presented for car E_2, which has a typical, modern high-requirement engine. The combined effects of coolant type and thermostat setting resulted in a change of 2.5 octane numbers. On a performance-number basis this effect is almost the same as that previously shown for car J.

Data were also obtained with three additional cars, and the combined effects of thermostat settings and coolants for the five cars are shown in Fig. 16. Going from the "summer" to the "winter" condition decreased the knock-limited spark advance an average of 5 deg on commercial fuel, while increasing the rating an average of one number. Concurrently, the engine requirement in terms of primary reference fuels was increased 4.5 octane numbers.

Combined Effect of Temperature, Humidity, and Cooling System Variables

With knowledge of the separate changes due to temperature, humidity, and cooling system variables, the next step was to investigate their combined effects. The results, for five cars, are shown in Fig. 17. Going from mild to severe conditions decreased knock-limited spark advance on commercial fuel by an average of 6.5 deg, while the ratings increased 2 octane numbers. The average change in engine requirement was 8.5 octane numbers, although individual vehicles varied from 4 to 15 units. Thus, as has been emphasized previously,

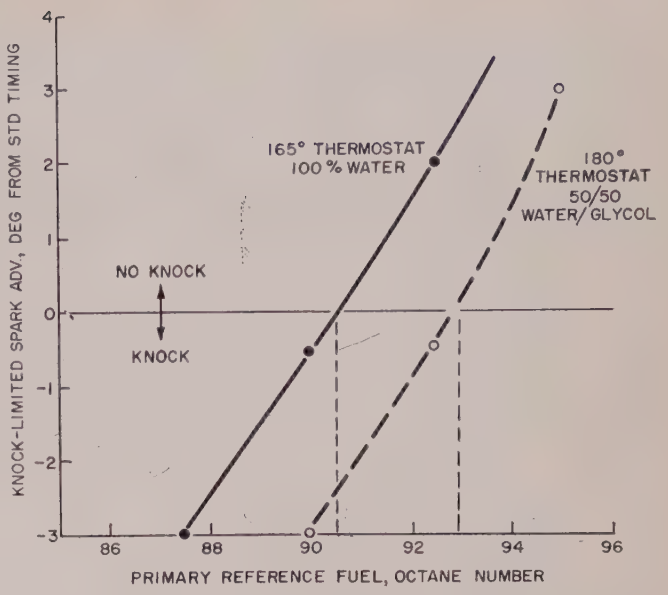


Fig. 15 – Effect of change in thermostat and coolant combination – temperature 50 F, humidity 31 grains per lb, car E_2

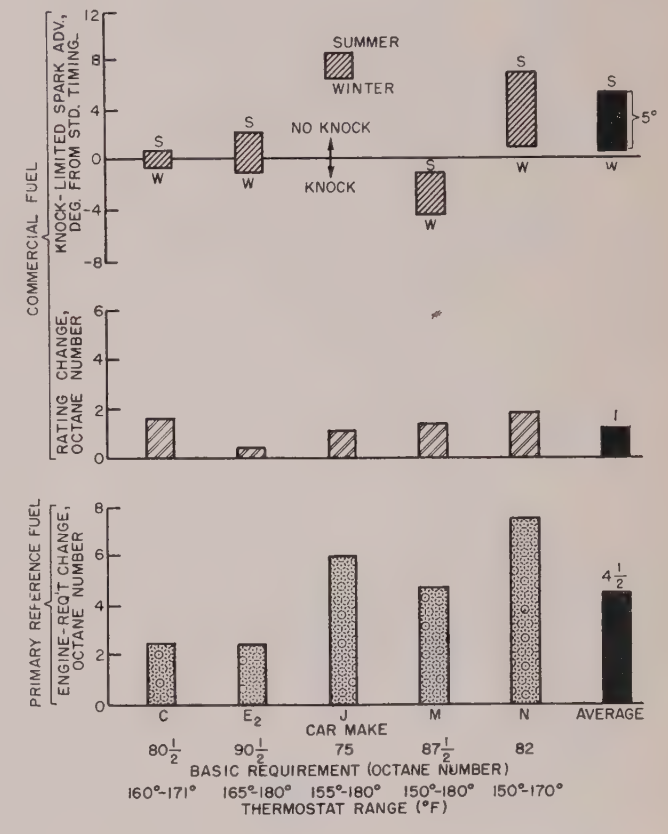


Fig. 16 – Effect of change in thermostat and coolant on fuel knock – temperature range 50-60 F, humidity 20-40 grains per lb, change from summer combination (low-temperature thermostat, water) to winter combination (high-temperature thermostat, 50/50 coolant)

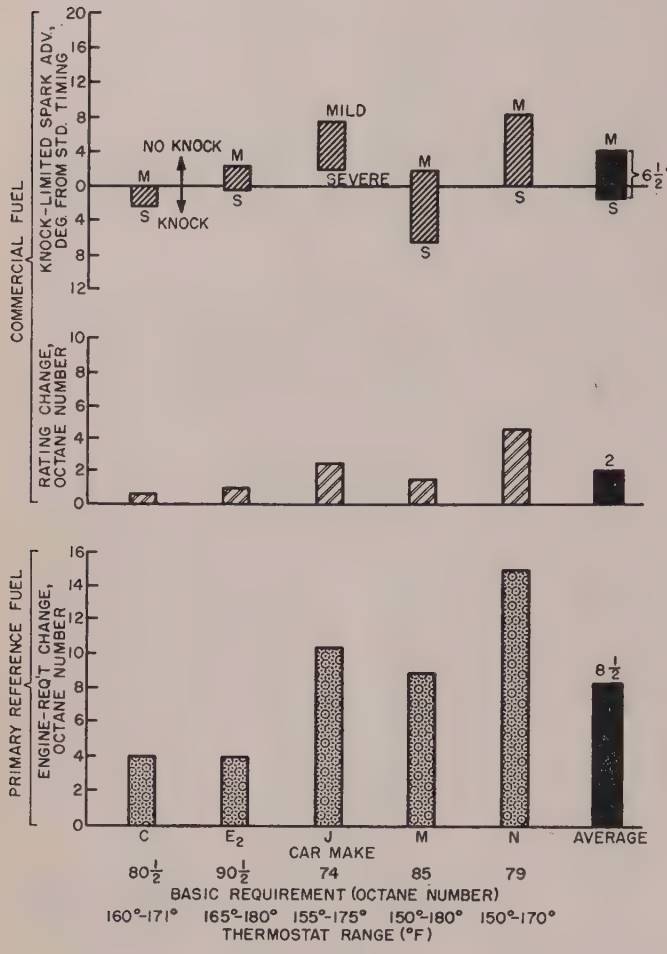


Fig. 17 – Combined effect of thermostat, coolant, and climatic extremes on fuel knock – change from mild conditions (50-60 F, high humidity, low-temperature thermostat, water coolant) to severe conditions (90 F, low humidity, high-temperature thermostat, 50/50 coolant)

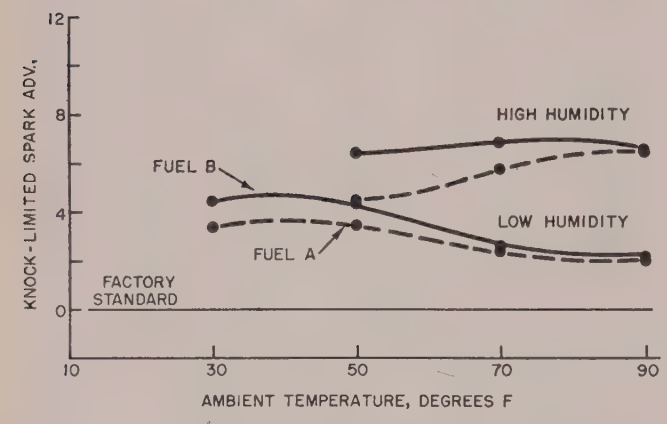


Fig. 18 – Effect of fuel composition on knock – high-temperature thermostat, 50/50 coolant, car J

while the fuel changes but little in rating, its ability to satisfy the vehicle changes in a marked fashion. Car M, for example, had a requirement level such that the changes due to the aforementioned variables took it from freedom from knock to severe knocking on commercial fuel.

Reid Vapor Pressure

An important fuel property to be investigated was Reid vapor pressure. It was thought that fuels of high vapor pressure might possibly cause some leaning out of the manifold air/fuel ratio at high atmospheric temperatures and induce knocking without vapor lock actually occurring. This was investigated by determining the effect of ambient temperature on knock-limited spark advance for 9.0, 10.5, and 12.0-psi test fuels over a temperature range from 30 to 90 F. Within the limits of reproducibility, the chassis-dynamometer results did not indicate any loss in vehicle octane-number rating with the 12.0- over the 9.0-psi Rvp fuel for the engine and climatic variables studied.

Effect of Fuel Composition

The data presented so far have been for average or typical fuels. Naturally, the possible effects of changes in fuel composition were of interest in this program. The effect of ambient temperature on the relative performance of two fuels of similar Research octane number and volatility specifications in car J is illustrated in Fig. 18. It will be observed that fuels A and B were equal at 90 F at both humidity levels, while fuel A became progressively poorer than B as ambient air temperature was reduced. Similar results were obtained when water instead of the 50/50 glycol-water mixture was used as the coolant. Inspection data indicated that the principal difference between fuels A and B was higher olefinic and lower aromatic content in the case of fuel A.

Two other comparable fuels were tested in car E_2, with the results shown in Fig. 19. The performance of fuels C and D differed with change in ambient temperature at both humidity levels. Fuel C was much higher in olefins and lower in aromatics than fuel D.

While the data presented are far from adequate to permit drawing conclusions, they do indicate that fuel composition offers a fertile field for investigation in the production of fuels that will exhibit minimum change in road antiknock satisfaction with change in atmospheric and cooling system variables.

Summary

A summary of the individual effects of atmospheric and engine cooling system variables is shown in Fig. 20. It should be emphasized that the

averages tend to mask the large effects observed in some car makes, and that these results do not necessarily represent the effects to be observed on all fuel-car combinations. It is important to keep in mind the conclusion that, while weather and associated variables change fuel octane-number ratings in some degree, they have a major effect on the ability of the fuel to satisfy the car requirement. Conditions requiring maximum fuel antiknock quality are found when combining the atmospheric effects of low humidity and high temperature with engine cooling effects of high-temperature thermostats and water-glycol antifreezes. Minimum requirements occur when the weather is cool and moist, water is used as the engine coolant, and a low-temperature thermostat is employed. Actual effects vary widely from car to car.

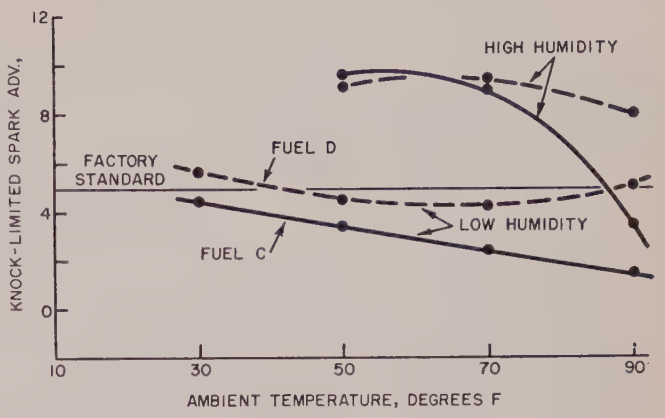


Fig. 19 – Effect of fuel composition on knock – high-temperature thermostat, 50/50 coolant, car E_2

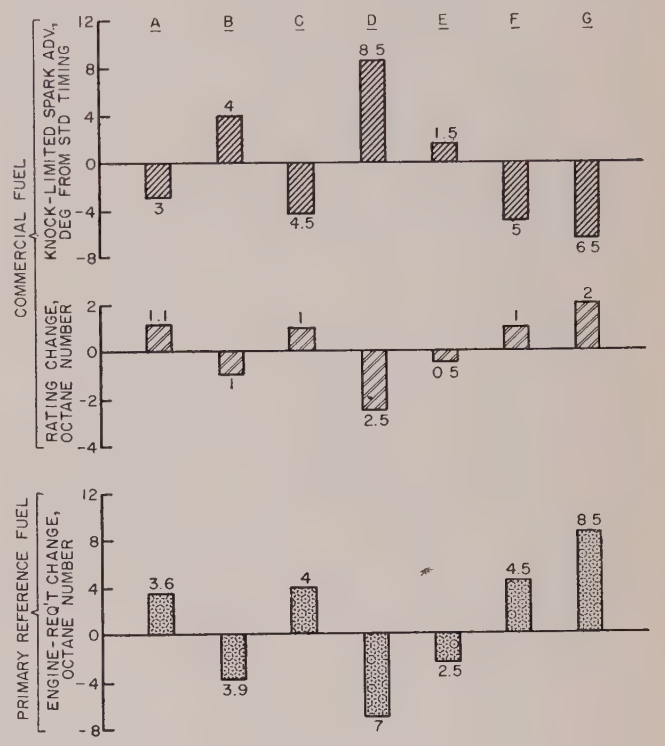
Significance

With the average variations as shown in Fig. 20, the question may well be raised as to what is the actual significance of this work to the automotive, petroleum, and chemical industries.

The automotive industry, in the past, has been occasionally criticized for building cars which exhibit a wide spread in octane-number requirements even when new. In one of the makes used in this study, weather and the associated variables changed the requirement by 15 octane numbers. This might well indicate that some of the wide variations in requirement reported in the past among cars of the same make and model were due to variations in test conditions rather than among the cars.

To the petroleum industry, this study indicates that weather and its associated variables are significant factors in fuel-rating reproducibility, and in the ability of fuels to satisfy customer cars. It also indicates that the study of fuel composition offers a fertile field for investigation in the tailoring of fuels that will exhibit the minimum change in road-antiknock value with changes in atmospheric and cooling system variables.

The chemical industry should be encouraged to develop improved permanent-type antifreezes. The work described indicates that in some cars there was an increase in engine requirement of several octane numbers accompanying the change from water to glycol-type antifreeze. This suggests an opportunity for progress in the development of antifreezes with improved heat-transfer characteristics. Until and unless this can be achieved, the practice of draining permanent antifreeze during the summer months should be encouraged.



A	B	C	D
Temperature	Humidity	Temp. & Humidity	Barometric Pressure
From: 30° To: 90° 7 Car Avg.	30 gr./lb. 130 gr./lb. 4 Car Avg.	50° F, High Hum. 90° F, Low Hum. 5 Car Avg.	30 in. Hg. 27 in. Hg. 2 Car Avg.

E	F	G
Wind Velocity	Thermostat & Coolant	Combined Effect of Temp, Humidity, Thermostat & Coolant
10 mph tail 10 mph head 3 Car Avg.	Summer Winter 5 Car Avg.	Mild Severe 5 Car Avg.

Fig. 20 – Average effect of atmospheric and engine variables on fuel knock

DISCUSSION ON FOLLOWING PAGE ——→

Combustion-Chamber Deposit Studies Must Consider Weather

– W. F. Ford
Continental Oil Co.

FOR years many of those in the business of suitably combining gasolines in engines for customer's service have known that weather conditions and altitudes have been influential factors on both engine requirements and fuel antiknock ratings. Selectively establishing octane numbers in specific sections of the country has more significance than just competition. To some perhaps the results of these authors explain periodic rashes of customers' complaints.

This study is an excellent approach to acquiring some finite understanding of the roles these outside factors play in satisfactorily combining engines and fuels. We only hope this is just the beginning of such activity and that further information will be developed on a wider scope.

Some interesting characteristics are presented in the paper. In Fig. 2 it appears that primary reference fuels and commercial fuels, as normally related in establishing either engine requirements of fuel antiknock or fuel antiknock ratings in cars on the road, do not maintain constant relationships. This plot indicates primary reference fuels to be relatively more sensitive to temperature increases than the commercial fuels included in the tests of the authors. This just reveals another complexity in the whole story of establishing engine requirements and fuel potentialities.

The straightline plots of primary reference fuels and commercial fuels at appreciably different slopes in Fig 6, where the effects of changes in humidity are presented, are interesting. We should like to know if these straightline characteristics are of significance.

In the last section of the report entitled "Significance," it was suggested that some of the wide variations in antiknock requirements among new or "clean" cars of the same make are caused by variations in weather and, accordingly, test conditions. No doubt there is some resultant effect from these changing conditions. Variations among cars of the same make still exist, however, when they are evaluated under relatively identical weather and operating conditions. It is felt that engines and fuels with even smaller variations can be developed. So far no one has been able to control or reduce changes in weather.

Recently, there has been increased interest and a greater amount of laboratory time devoted to the subject of combustion-chamber deposits. Even in these studies weather factors add confusion to an already very complicated situation. Any conclusions drawn on the effects of fuels, lubricants, engine designs, and operating conditions in the formation of combustion-chamber deposits and their resultant effects on antiknock and preignition requirements must take into account these weather variables.

An indication of the magnitude of the effect of weather variables on combustion-chamber deposits and a requirement test result reproducibility was indicated in a report of limited distribution by one of this paper's coauthors, Mr. Gibson. Excerpts from his report, written in 1947, are:

In the "summary" section: "Reproducibility of tests designed to study the effect of fuels and lubricants on deposits and the effect of the deposits on requirements is poor in some engines and apparently relatively good in others. This seems to be affected to a large degree by atmospheric temperature and humidity at the time of test observation. Possible variations in temperature and humidity change the octane-number requirements of some engines by as much as the increase in octane-number requirement caused by deposits in the engine."

In the "discussion" section: "Atmospheric variables are important in that they are known to have considerable effect on the requirement of a given vehicle and because there are definite indications that the effect may be much different with a clean engine and with deposits. For example, in one popular car a change in moisture content from 25 to 100 grains per lb of dry air changed the clean-engine requirement by 6 units, while a corresponding change with deposit produced only a 2-unit change. Another popular car showed about half this effect due to humidity and was also temperature sensitive. Still another car had a compensating effect from temperature and humidity. This particular make of car has shown a high degree of reproducibility of data in road tests. Single-cylinder engine data and such car data as are available indicate a change of 2 to 3 units per 1-in. change in barometric pressure. Obviously, if all these factors should add up, requirement change reported as due to deposits might easily be doubled or halved as the case might be. This is a convincing argument for a statistical approach to this problem."

In the work of the various groups and panels of the CFR Motor Fuels Division on octane requirements and ratings of fuels it is realized that the factors considered in this paper can seriously affect final results. Some of the participants in these activities feel a cooperative program in the nature of the one in these papers should be conducted.

Data Correlate Well With Other Data

– J. P. Hamer
Esso Laboratories

IT may be of interest to present the correction factors which we have used, although these are not based in general on as extensive a study as that outlined in this paper. The generally excellent agreement between our information and the data presented in this paper is shown in Table A. (The same ranges of conditions as are used in expressing the results in the paper are also applicable to our information.)

In connection with rating fuels at various elevations, we have observed an average decrease in commercial fuel ratings of 1.6 octane numbers in going from sea level to 6500 ft above sea level. In comparison, the authors show an average decrease in the rating of commercial fuels of about 1.5 units for a 6000-ft elevation. In this connection, it has been demonstrated by the Bureau of Standards that octane-number corrections for barometric pressure are very greatly influenced by the level of octane requirement being considered. For example, a 6-unit decrease in octane-number requirement for an increase of 3000-ft elevation was observed at a level of approximately 83 road octane number at sea level. However, at a level of 70 road octane number, the decrease in requirement would be expected to be about 12 units at the same elevation. It may well be that the other weather conditions may also influence octane requirements as a function of the requirement level. For this reason, the corrections discussed should probably be applied only in the octane range of current commercial gasolines and requirement range of present cars.

All of the operating conditions and weather factors discussed in this paper appear to be largely applicable to relatively low-speed knock. When considering high-speed knock, it is believed that the corrections in requirement will be materially reduced.

Table A – Engine Octane Requirement Change
(Primary Reference Fuels)

	"Weather or Knock"	Esso Laboratories
Temperature	+3.6	+4.5
Humidity	−3.9	−4.8
Temperature Humidity	+4.0	+4.0
Barometric Pressure	−7.0	−6.0
Thermostat and Coolant	+4.5	+4.0
Combined Effects	+8.5	+8.0

Table B – Effect of Altitude on Octane Requirement – Comparison Between 1949 CRC Survey and 1949 Phillips-Gulf Survey at Amarillo, Tex.

Car Make	Average Decrease in Octane Requirement: For 3000-Ft Increase in Altitude	Average Decrease in Octane Requirement: Per 1000-Ft Increase in Altitude
L	4.6	1.5
G	6.4	2.1
N	5.8	1.9
I	10.8	3.6
T	8.0	2.7
O & Q	14.8	4.9
Average	8.4	2.8

Consider "Hill Factor" Effects On Engine Octane Requirements

– J. E. Taylor
Gulf Research & Development Co.

I WOULD like to present some additional data which, in general, agree with the data reported by the authors. The octane requirement data to be presented are maximum requirements obtained in a manner similar to that employed by the authors.

Fig. A shows the effect of altitude on the octane requirements of three car makes. This work was done several years ago in the vicinity of Pittsburgh, Pa., where an altitude change of 1500 ft is readily available. Atmospheric temperatures and humidity were essentially the same at both elevations. All three cars contained overhead-valve engines, one was a V-8, another an in-line 8, and the third a 6.

The data show that the decrease in maximum requirement per 1000-ft increase in altitude was similar for these three makes, averaging about 3.5 octane numbers in terms of primary reference fuels and about 2.5 octane numbers in terms of the Research Method rating of commercial gasolines. This means that the rating of the commercial gasolines was reduced about one octane number when the test altitude was increased 1000 ft. These data are in good agreement with the data reported by the authors.

Table B presents a comparison between the octane requirements determined for a group of cars in the 1949 CRC Octane Requirement Survey and the requirements of the same car makes determined at Amarillo, Tex. during the summer of 1949 in a special survey conducted jointly by the Phillips Petroleum Co. and the Gulf Research & Development Co. The purpose of this survey was to determine whether the effect of barometric pressure on octane requirements was actually reflected in the requirements of customer cars normally operated at moderate altitudes. The altitude at Amarillo, Tex. is 3660 ft, which we figured to be about 3000 ft higher than the altitude for the CRC survey based on the assumption that the average altitude of laboratories participating in the CRC survey was 600 ft.

The data show that the maximum requirements of all car makes was significantly lower at Amarillo, Tex. than they were at lower altitudes. The average value of 2.8 octane numbers per 1000 ft is in good agreement with the data reported by the authors. From these results, we concluded that the effect of altitude on the octane requirement of cars in normal customer use is similar to that which would be predicted from engineering studies. In general, the carburetion of these cars had not been modified and the ignition timing was quite normal.

Fig. B presents the results of some tests that we conducted in 1939 to determine the effect of various antifreeze materials on octane requirements. As shown in Fig. B, a very significant effect was observed. Of particular significance is the fact that the alcohol type of antifreeze materials increased requirements about one-half as much as the glycol type. It is also of interest to note that the effect of methyl and ethyl alcohol is essentially the same. Although these numbers appear very large, the basic requirement of this engine was only 71-octane-number fuel and if these increases were put on the basis of performance

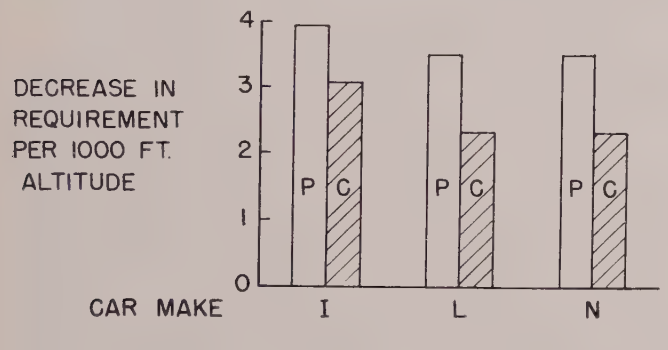


Fig. A – Effect of altitude on octane requirement
P – Primary reference fuels
C – Commercial gasolines, Research rating

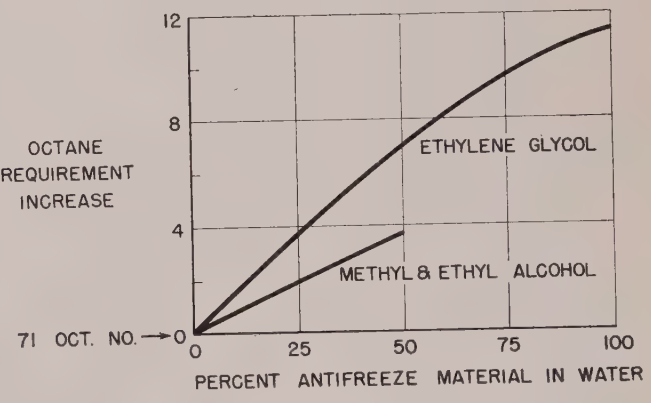


Fig. B – Octane requirement increase due to antifreeze materials – 1939 car, make N

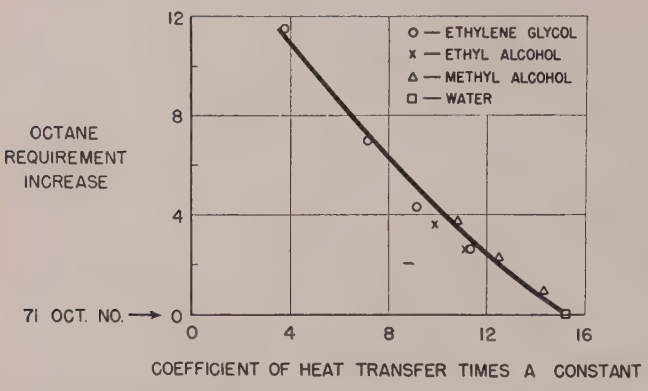


Fig. C – Relation between octane requirement increase and coefficient of heat transfer for various antifreeze materials – 1939 car, make N

numbers to consider the nonlinearity of the octane scale, the values shown would be divided by approximately 2.5 to convert them to the 90-octane-number level.

Fig. C shows the results of an effort to rationalize the rather substantial increases in octane requirement presented in Fig. B. This plot shows the relationship between octane requirement increase and a function of the coefficient of heat transfer. The constant referred to accounts for the size of the coolant passages and the coolant velocity, which are unknowns but the same for the various coolant mixtures. The fact that the data points for the 10 different coolant mixtures define a rather definite line indicates quite strongly that there is a real relationship between octane requirement increase and the coefficient of heat transfer.

Fig. D presents some data to show that there is another very important factor that controls the octane requirements of engines. This is the "hill factor," which is simply the difference between requirements obtained under severe operating conditions on steep hills and those obtained under relatively mild operating conditions on level road. In Fig. D the effect of hill climbing varies from a minimum of a little over one octane number for car B to nearly 4.5 octane numbers for cars N and I, with an average of about 2.5 octane numbers. It is of interest to observe that the

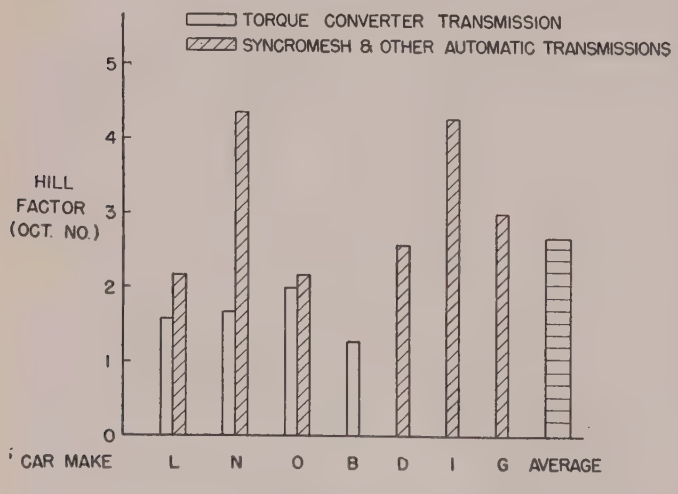


Fig. D – Effect of hill climbing on octane requirement – 1949-1951 passenger cars

cars equipped with torque converter transmissions typically had lower hill factors than cars equipped with synchromesh or other automatic transmissions that permitted sustained full-throttle operation at low engine speeds. We believe the effect of hill climbing on octane requirements is pertinent to this discussion for two reasons: (1) In some cars, its effect is greater than the effect due to practical variations in atmospheric conditions; and (2) atmospheric correction factors based on level road requirements may not be directly applicable to hill requirements.

In the data presented in the paper, the effect of ambient air or jacket temperatures on the octane rating of fuels appears to be in the opposite direction to that normally expected. In Figs. 2, 3, 4, 16, 17, and 20, higher operating temperatures increased the rating of commercial and other sensitive fuels, whereas, it is generally believed that high temperature depreciates the rating of sensitive fuels. Also, in Fig. 3, a leaded straight run is depreciated by temperature more than any of the sensitive fuels tested. We wonder if the authors have any explanation for the difference between their test results and the generally accepted effect of temperature on fuel ratings.

The determination of accurate octane requirements plays such an important part in the research programs that many automotive and petroleum laboratories are conducting today that I wonder if consideration should not be given to the development of a cooperative program to establish firmly the effects of atmospheric variables on the octane requirements of all current-model engines.

Several Questions For the Authors

– L. J. Grunder
Richfield Oil Co.

ONE of the more important statements made in this paper is: The importance of relating fuel octane-number ratings to engine requirement must be made at the same temperature if the ability of a fuel to satisfy the engine in which it is being used is to be completely evaluated.

What this statement says is that the road test crews apparently found that the temperature, humidity, and wind velocity and direction changed between the time of rating the reference fuels and the experimental fuels, and that the dynamometer tests were run indoors under conditions held constant for an antiknock determination.

On the road the driver and observer usually operate the car with windows closed and on a level road. In the tests run on the dynamometer, were the rates of acceleration matched against road acceleration for each of the nine cars tested, and was the trace knock level the same for both types of road tests – indoors and outdoors?

Some years ago, in attempting to control hot-weather coolant temperatures, we had occasion to check coolant pressures and radiator cap temperature control devices. We found extremely poor repeatability of high-speed, hot-weather road ratings if radiator caps of the same markings were interchanged. We also found out why numbers are not shown on the dash gage for water temperature.

Nothing was mentioned about manifold heat deflectors that stick in the open position, or partially open position. A casual parking lot check on five cars showed four stuck, and three cars were later models than the cars chosen for the test report of these authors.

Regarding the selection of three cars – because of field experience indicating they were particularly sensitive to seasonal changes, were these cars N, J, and M?

In the test fuel analyses, especially the temperature, humidity, and coolant tests, commercial fuels – their Reid vapor pressures, 10% and 50% points and gravities look very much like winter grades. However, the six base

stocks used for the temperature, humidity, and wind velocity tests – on the dynamometer – appear to be summer grades.

With winter fuels used during the dynamometer coolant tests, which simulated winter driving conditions, how then does it correlate with data when using plain water as coolant in summer and summer-grade fuel?

The authors state that the Modified Uniontown Method Technique is used, which method gives basic spark timing to give an audible trace knock intensity. Would the authors clarify the statement made later, to the effect that at the recommended basic timing there is zero knock.

Our road tests show some 2 to 5 deg of retard is necessary in order to eliminate a trace knock down to zero knock, and this varies from car to car. My question is – how do the authors take care of that difference on their charts?

Lastly, was the distributor calibrated to factory standards before these tests were run, or was the car accepted "as is" from the owners?

Additional Test Results Confirm Authors' Data

– R. C. Buck
GM Research Division

WHETHER or not an engine will knock with a given fuel depends on the octane rating of the fuel relative to the octane requirement of the engine. In order to realize the maximum benefits obtainable with higher compression ratios, automotive engineers strive to develop engines that make the maximum utilization of the available commercial gasolines. As a result, the amount by which the fuel ratings exceed the engine requirements, in cars giving satisfactory knock-free performance, is often only one or two octane numbers. For this reason it is important that we know the extent to which weather changes affect fuel octane ratings and engine octane requirements in cars on the road.

To obtain a better understanding of the effects of temperature, pressure, and humidity on knock measurements, a series of 38 separate road tests were made using a 1952 Chevrolet equipped with synchromesh transmission. These tests were conducted at the GM Proving Ground. In each test, engine octane requirement was determined and three commercial fuels were rated by means of the CRC Modified Borderline Technique. All of the tests were conducted by the same team during a 6-month period from June to December. Weather conditions existing during each test were recorded. During the test period atmospheric conditions ranged between the following limits:

Barometer, in. Hg	29.49 – 29.15
Temperature, F	32 – 87
Humidity, grains water per lb dry air	13 – 87

Combustion-chamber deposits at the beginning of the test represented 2000 miles of high-speed driving. All subsequent driving of the car was limited to the test operations. This consisted of a 10-mile warmup at top speed followed by approximately 20 full throttle accelerations between 800 and 3000 rpm before each rating. It was felt that this procedure would keep variations in deposit conditions at a minimum. A statistical analysis of the data from these tests was made by Dr. L. Johnson of our engineering mechanics department to determine the extent to which the octane-number requirement and the octane-number rating of the fuels were dependent on the temperature, pressure, and humidity. A linear equation was developed for each speed, which described the octane-number requirement in terms of these variables. Similar equations relating the octane-number ratings of the three fuels were developed. The results, which agree with the findings of these authors, show that the octane requirement of the engine increases with increasing temperature and pressure and decreases with increasing humidity. The octane ratings of the fuels remain essentially unchanged.

Table C – Change in Octane-Number Requirement with Changes in Weather

	Change in Octane-Number Requirement with		
	1-In. Increase in Barometer	60 F Increase in Temperature	100 Grain per Lb Increase in Humidity
800 Rpm	+3.02	+4.48	−4.27
1000 Rpm	+3.31	+4.19	−3.60
1200 Rpm	+2.61	+4.21	−3.61
1600 Rpm	+2.31	+4.08	−5.43
2000 Rpm	+1.71	+1.83	−4.94
Average	+2.59	+3.76	−4.29
Potter, Scott, Gibson, and Stanke	+2.33	+3.6	−4.0

Table C shows the effect of the weather variables on the octane requirement of the engine at each of the five speeds, as determined by our analysis. Included at the bottom are the changes reported by the authors. The figures shown are the changes in octane-number requirement occurring with a 1-in. change in barometer, a 60 F change in temperature, and a 100 grain per lb change in humidity. The results presented by the authors of the paper were for a 3-in. change in barometer, and for this comparison we have simply divided their results by three. We encountered a total change in barometer of slightly less than 1-in. and are not sure that our correlation factor would be the same over the greater range.

Their figures are the average changes at the speed of maximum requirement, which varies somewhat with the car. In our tests the maximum requirement was observed at 800 rpm in every case. The coefficients from our results at 800 rpm are slightly greater than the average obtained by the authors, while our coefficients, averaged for all speeds, are in excellent agreement with their coefficients. It is interesting to note that our coefficients are speed sensitive. The numerical value of the pressure and temperature coefficients decrease as the speed increases. The value of the humidity coefficients increase slightly with increased speed.

In Table D, the effects of the weather variables on the average octane ratings of the three commercial fuels are shown. Included at the bottom are average changes reported by the authors of the paper. Again we have compared the changes occurring with a 1-in. change in barometer.

We have found that the relationship between fuel ratings and weather differs for different fuels. The figures shown are the average changes in octane rating of the three commercial fuels at each speed. Our coefficients averaged for all speeds show a 0.43 decrease in octane rating with a 1-in. increase in barometer, a 0.68 decrease in rating with a 60 F increase in temperature, and a 0.39 decrease in rating with a 100 grain per lb increase in humidity. Almost all our coefficients are small, indicating very little dependence of fuel rating on the weather variables within the ranges studied.

The effect of temperature and pressure was opposite to the effects observed by the authors. Our fuel ratings decreased slightly with increases in temperature and pressure, while their ratings increased slightly.

In Table E, we have compared the effects of the weather variables on the borderline knock spark advances of the commercial fuels. The figures are the average change in borderline spark advance for the three commercial fuels. In this comparison we agree closely with the results of

Table D – Change in Fuel Octane-Number Rating with Changes in Weather

	Change in Octane-Number Rating per		
	1-In. Increase in Barometer	60 F Increase in Temperature	100 Grain per Lb Increase in Humidity
800 Rpm	−0.32	−0.27	+0.10
1000 Rpm	−0.52	−0.48	+0.32
1200 Rpm	−0.54	−0.30	−0.44
1600 Rpm	−0.61	−1.08	−0.56
2000 Rpm	−0.12	−1.28	−1.35
Average	−0.43	−0.68	−0.39
Potter, Scott, Gibson, and Stanke	+0.83	+1.1	−1.0

Table E – Change in Borderline Knock Spark Advance with Changes in Weather

	Change in Borderline Knock Spark Advance per		
	1-In. Increase in Barometer	60 F Increase in Temperature	100 Grain per Lb Increase in Humidity
800 Rpm	−3.81	−2.25	+3.12
1000 Rpm	−4.17	−1.90	+3.19
1200 Rpm	−3.46	−1.48	+3.7
1600 Rpm	−3.97	−1.32	+4.7
2000 Rpm	−3.42	−1.52	+4.6
Average	−3.77	−1.69	+3.86
Potter, Scott, Gibson, and Stanke	−2.83	−3.0	+4.0

the authors. They observed approximately a 3-deg change per 1-in. barometer change, while our average change was approximately 4 deg. For a 60-deg change in temperature they obtained a 3-deg change, while our average change was approximately 1½ deg. For changes in humidity we both obtained approximately a 4-deg. change.

In Fig. E we see the results of our attempts to apply a weather correction factor to the observed results. At 800 rpm the maximum requirement observed for all the tests was 86 octane numbers, while the minimum was 78 octane numbers – a spread of 8 octane numbers. The observed requirements were corrected to the mean weather conditions by means of the equation describing the requirement at 800 rpm. These mean weather conditions were: barometer 28.85 in. Hg., temperature 56 F, and humidity 46.5 grains water per lb dry air. This reduced the maximum to 85.5 and raised the minimum to 79.5 – a spread of 6 octane numbers. It was concluded that the remaining spread of 6 octane numbers was due to errors in measurements and other unknown factors. Since the remaining spread amounted to 75% of the original, we believe that there is considerable work yet to be done on the development of correction factors.

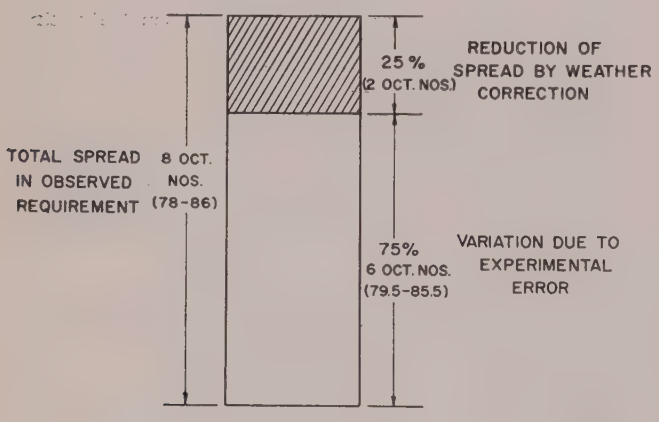


Fig. E – Result of attempt to apply weather correction factor to observed results

Questions Authors' Conclusions Concerning Ethylene Glycol

– F. A. Gundlach
National Carbon Co.

AS a representative of the antifreeze industry, we are confining our comments to the section of the paper devoted to thermostats and high-boiling-point antifreeze and their effect on knocking tendencies. As a basis for the discussion, two very important points must be kept in mind:

1. Water is unsatisfactory as an engine coolant during winter weather. Properly inhibited antifreeze must be used. A higher boiling point than water may also be desirable to reduce boiling at hot spots causing steam blanketing, which really raises local temperatures.

2. Other authorities have indicated that high-temperature thermostats and other suitable temperature-control devices are essential to maintain proper car performance during cold weather: reducing warmup time, reducing sludge tendencies due to condensation of unburned fuel or water in the crankcase, and cutting down general engine wear due to low-temperature operation conditions.

We have no doubt that the data presented in this paper accurately portrays the conditions found in chasis dynamometer work, where the variables are studied independently. However, we do question the group of cars selected as being representative of the mass of cars on the road today. In addition, we take serious exception to the sweeping conclusions in the abstract, which are not related to conditions in cars as they are driven by the motoring public.

Previous papers given by one of the authors have indicated that combustion-chamber deposits from either fuels or lubricants, as well as driving conditions or performance characteristics of either the fuel induction or ignition systems all affect octane requirements to such an extent that thermostat or coolant variables are insignificant.

This has been verified by our own closely controlled road test work with water, alcohol, and glycol solutions. Octane demand curves run with these three materials over a wide operating range overlapped each other and all fell within a very narrow band. We feel that this range was within the limits of overall engine performance reproducibility on the road.

As regards use of ethylene glycol in normal antifreeze concentrations, an investigator of high-temperature cooling in high-output aircraft engines during the last war reported that 50% water solutions were as effective as water alone in regard to knocking tendencies.

We believe that proper evaluation of knocking tendencies must be based on changing conditions in the combustion system and cooling system – rather than pin-pointing fuel, air, or coolant variables. For instance, cooling system deterioration in service, particularly deposit formation, can have a considerably greater effect on knocking tendencies than variations in thermostat-opening temperatures or currently accepted coolants. We are wondering if these factors have been considered and evaluated by the authors in their extensive work.

Research on improved coolants is being continuously carried on. Other requirements of coolants besides heat transfer, such as freezing point depression, corrosion protection, defoaming, and boiling point, must be considered. Our recommendations for replacing coolant in the spring and fall have been based on the need for inhibitor replenishment to protect against rust deposits, which can affect cooling system performance and thus increase octane requirement to a much greater degree than that reported for a change from water to glycol-base antifreeze.

The New Packard Lightweight Diesel Engines

M. Ware, R. E. Taylor and J. Witzky, Packard Motor Car Co.

Paper presented at SAE National Diesel-Engine Meeting, Chicago, Nov. 4, 1953.

THE new family of diesel engines developed by Packard provides reductions in weight and space occupied, while maintaining good life expectancy. Previous Packard experience in developing and building lightweight high-output engines dates back to before World War I, when Packard had a prominent part in the Liberty engine program. This was followed by a series of watercooled aircraft engines, which contributed to development progress after the war. Later on there was the pioneer experience with the lightweight diesel aircraft engine. Prior to and during World War II a series of high-output liquid-cooled engines was developed for use in PT and air-sea rescue boats. During the wartime production of Merlin engines, concurrent development programs resulted in a 70% increase in the authorized rating of the engine.

As the development of the engine for the PT boats progressed, life was increased substantially not only for extended operation at emergency ratings but also for total hours at cruising ratings. These more recent experiences were of importance in the present diesel program in contributing definite guides for accomplishing the desired type of diesel engine with relatively long life potential.

A family of four engine sizes was created – 6′, 8′, 12′, and 16-cyl basic models – all four stroke. The 6-cyl engine is an in-line unit, the others are 60-deg V engines. They have a large number of interchangeable parts and common features. The bore and stroke are the same for all engines.

There have been a number of lightweight diesels that are lighter than the new Packard engine. These are primarily aircraft types and include those construction features generally associated with aircraft engines. Aircraft-engine ratings usually involve a take-off or emergency rating that

THE new Packard automotive diesel engine is reported to feature reductions in weight and space occupied, while maintaining good life expectancy.

It is available in 6-cyl in-line, V8, V12, and V16 models. These models all have a large number of interchangeable parts. They all have the same bore and stroke.

According to the authors, noteworthy points of these engines include:

1. **Extensive use of aluminum.**
2. **Turbosupercharging.**
3. **One-piece cylinder and head assembly.**
4. **Four valves per cylinder, with stellite seating surfaces on valves and seats in the head.**

The Authors

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R. E. TAYLOR is assistant chief engineer, Diesel and Industrial Engineering Activity, Packard Motor Car Co. Mr. Taylor worked four years at Chrysler and did engine research at the University of Kentucky before joining Packard in 1941.

J. WITZKY (M '50) is a project engineer for the Packard Motor Car Co. He was born and educated in Stuttgart, Germany. Mr. Witzky joined Packard in 1948 as a development engineer.

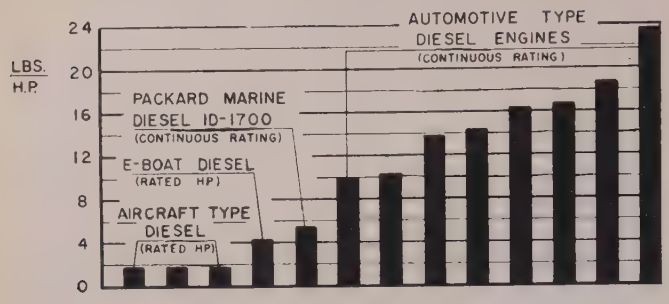


Fig. 1 – Weight per horsepower for various diesel engines

is considerably in excess of authorized cruising ratings for extended operation. This type of rating is properly reflected in the type of engine structure.

Weight per hp for these engines and for representative automotive-type diesel engines is shown in Fig. 1. For the automotive-type engines, weights as related to continuous rated horsepower are shown, as obtained from the statistical issue of *Automotive Industries*. The Packard engines are intermediate in position as compared with the two classifications.

Preliminary Development

A sample 6-cyl engine, constructed especially for combination with a high-speed generator to provide a very lightweight combination unit, was designed, built, and tested. This engine demonstrated the practicability of the overall design features. Experimental sample cylinders were made at the same time and combustion-chamber relations were investigated to evolve a favorable combination. A single-cylinder engine was used for this purpose. In proceeding from these experiences to the present engines, stress analyses and design investigations were based on higher horsepower ratings to be sure that the engines would have long life at the continuous ratings and also that there would be ample structures for higher ratings as future applications might require. The 6-cyl and 12-cyl models were started almost simultaneously and the requirements also involved two different hydraulically operated reverse gears, one with a coaxial 3.2/1 reduction gear and the other with an offset 5/1 reduction gear. Generator sets with lightweight bases were also designed.

Engine Features

Noteworthy features of the engine include:

1. Extensive use of aluminum in order to obtain weight reduction.
2. A substantial crankshaft with considerable excess in bearing capacity for long service life.
3. Injection of fuel into a precombustion chamber for the purpose of eliminating high peak piston loading pressures.
4. Turbosupercharger providing exhaust heat recovery.
5. Hard-surface cylinders to reduce wear and minimize service replacements.
6. Nitrided crankshafts, to provide maximum endurance strength and maximum journal and crankpin life.
7. Four valves per cylinder with stellite seating surfaces on valves and seats in the head.
8. One-piece cylinder and head assemblies, eliminating problems in sealing cylinder gas pressures.
9. Gear drives for all accessories with all gearing located at the flywheel end.

The general objectives are obtained by providing moderate supercharging for continuous operation at moderate piston speeds. Rated power is obtained with 140-psi bmep and 2100 fpm piston speed.

The primary outside dimensions of the 12-cyl, 600-hp engine (1D-1700) are shown in Fig. 2. The relative space occupied, as compared with a much

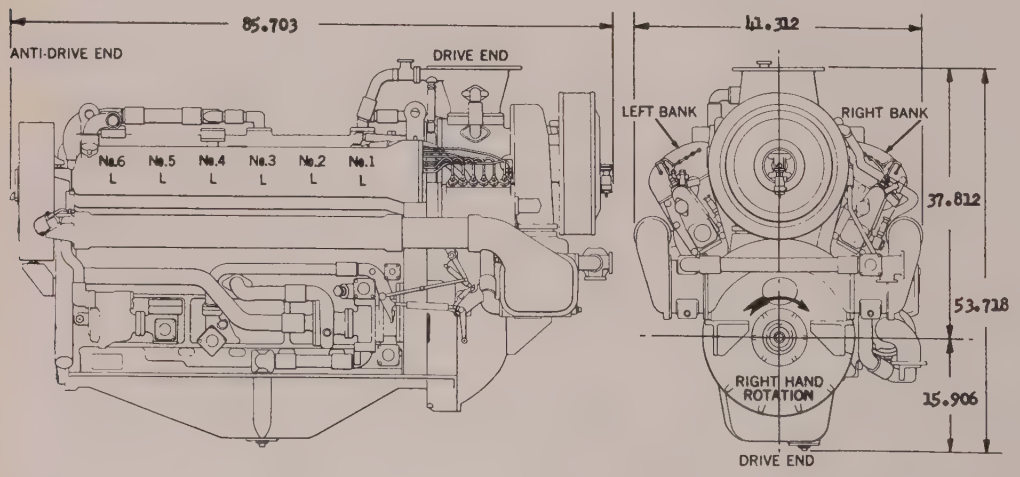


Fig. 2 – Packard 12-cyl, 600-hp engine

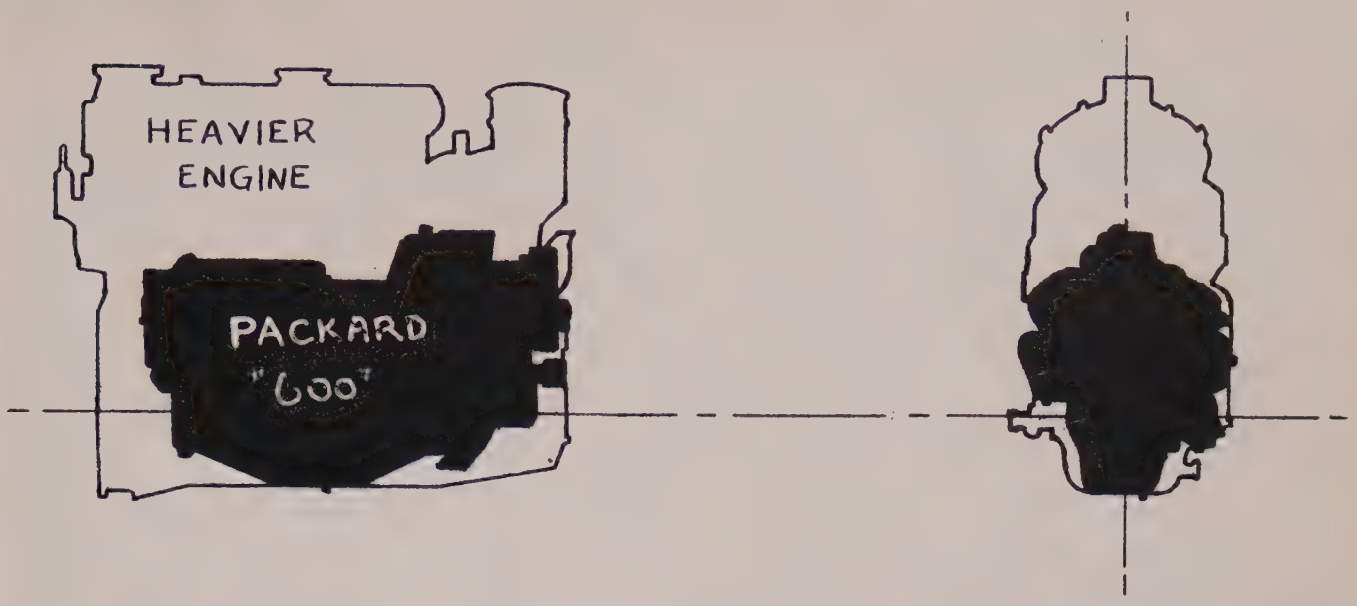


Fig. 3 – Relative space occupied by Packard 12-cyl engine and much heavier engine of same rating

heavier engine of the same rating, is shown in Fig. 3. Fig. 4 shows a comparison with a slow-speed engine having the same output. This engine weighs more than 35,000 lb, which is in excess of 10 times the weight of the 1D-1700 engine.

Basic engine specifications are shown in Table 1.

Construction Details

Fig. 5 shows a longitudinal section of the 12-cyl engine. Fig. 6 shows a transverse section.

The cylinder block and crankcase are formed as an integral aluminum casting, made of 355 aluminum alloy with overaged heat-treatment. This alloy and heat-treatment provide a relatively high endurance limit. It is the main part of the overall engine structure and, by virtue of the integral construction and design detail, provides rigidity, which contributes considerably to the long life characteristics of the engine. A steel tube is cast into the

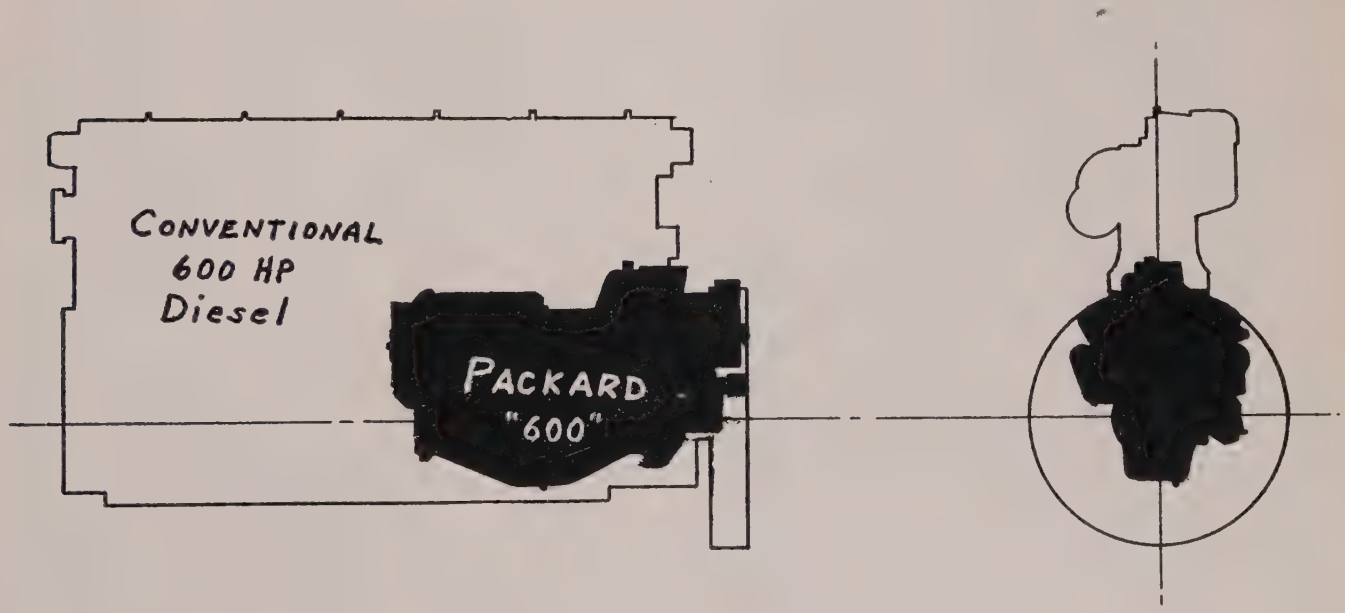


Fig. 4 – Relative space occupied by Packard 12-cyl marine engine and slow-speed engine of same output

Table 1 – Basic Engine Specifications

	6-Cyl Engine	12-Cyl Engine
Model	2D-850	1D-1700
Continuous Rating, hp at 2000 rpm	300	600
Bore and Stroke, in.	5⅜ x 6¼	5⅜ x 6¼
Piston Displacement, cu in.	852	1704
Compression Ratio	15/1	15/1
Valve Arrangement	Overhead (4 per cylinder)	
Combustion System	Precombustion chamber	
Crankshaft	Steel forging – surface hardened	
Lubrication System	Dry sump	
Starting	Compressed air or electric motor	
Fuel System	Multiplunger pump with pressure-operated injection nozzles	
Cooling System Pump, gpm	100	200
Supercharger	Exhaust gas turbine driven centrifugal compressor	
Basic Engine Weight, lb	2195	3300

crankcase forming the main lubricating oil gallery. The bearing caps are aluminum forgings. Long bolts extend from side to side of the crankcase and pass through all bearing caps, completing an approach to a solid structure completely surrounding the main bearings. It may be of interest to note that this 12-cyl block and crankcase casting weighs 550 lb after machining.

Combining a substantial crankshaft with the rigid crankcase and good bearings is a major contribution to the long life objective. These three elements together form the backbone of the engine. The journals are 4 in. in diameter, the crankpins, 3⅝ in. The cheeks are 1-in. thick. Integral balance weights provide 58% balance. Crankpin bearing loading is shown in Fig. 7; loading on the center main bearing, which is the maximum journal loading, is shown in Fig. 8. These figures show that bearing capacity has not been sacrificed for light weight.

The crankshaft is nitrided all over. This not only provides outstandingly hard journal surfaces for wear resistance but also gives about the maximum possible fatigue strength using a commercial material.

The unit cylinder and head assembly is shown in Fig. 9. This construction, while possibly more expensive than the separate sleeve and removable head construction usually specified, eliminates the troubles that are always re-occurring with high-pressure cylinder-head gaskets. The inside diameter of the cylinder is nitrided in order to provide a hard wearing surface and thus extend the life of the cylinder and minimize servicing requirements. The extended life also helps to compensate for any greater costs of the combined construction.

The valve seating surfaces on both the cylinder heads and the valves are stellited to provide, again, long life between service operations.

Inverted cup pearlitic malleable cam followers operate in cylindrical guides cast integrally with the valve guides. There are two camshafts for each cylinder bank.

The cylinder and head assembly is formed by combining a 4130 precision cast cylinder head with a nitralloy barrel welded to a short extension formed on the head. Water spaces in the head are completed by two welded plates. All welds are done

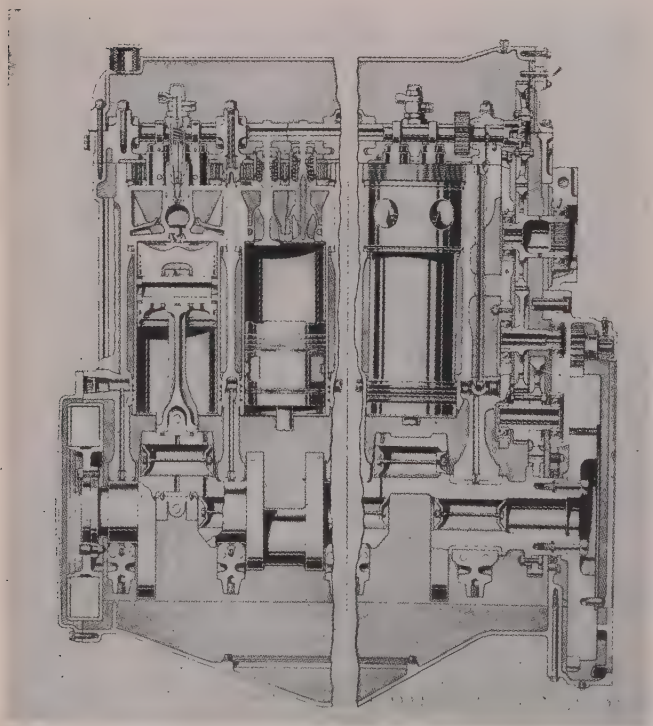


Fig. 5 – Longitudinal section of 12-cyl engine

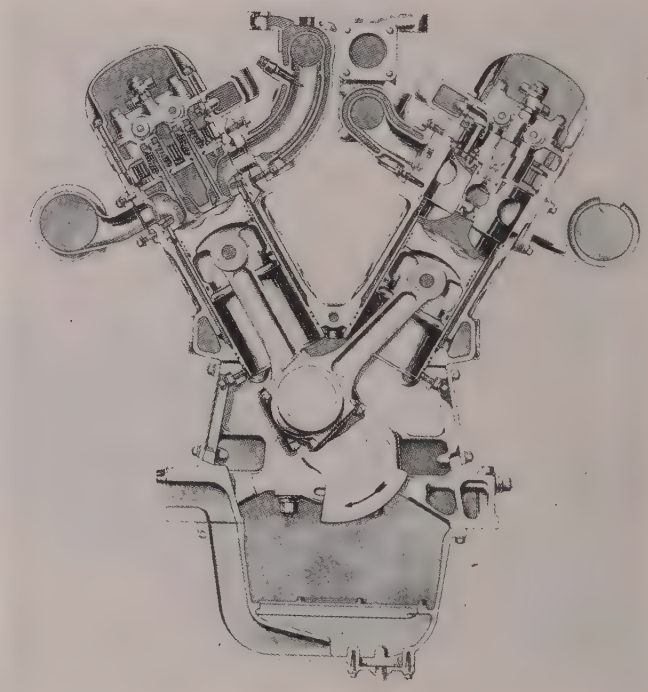


Fig. 6 – Transverse section of 12-cyl engine

by machine, providing a uniformly dependable structure.

Precision casting of the cylinder head contributes several important features.

1. Accurate control of cast dimensions permits reduced wall thicknesses.

2. The surfaces of the port passages are relatively smooth in the cast form, so that very little hand finishing is required.

3. The precombustion chamber does not require machining.

4. The precision casting method helps to provide a structure especially arranged to control the cooling water flow through the head.

The water enters the head through a series of holes located at the periphery of the cylinder head, and communicating with the jacket space around the barrel. A baffle within the head directs all the water substantially radially inward to the precombustion-chamber location, where the water rises to enter a space in the top of the head. In this way, all of the water is utilized in cooling the head, the precombustion chamber, and the injection nozzle.

The fuel is injected directly into the precombustion chamber, which communicates through a series of orifices with a toroidal space over the piston. This type of combustion chamber was chosen because of its ability to provide high mean effective pressures without high cylinder pressures and with moderate fuel line pressures. Both the low cylinder pressures and the low fuel pressures are long service attributes.

The piston is an aluminum forging. Five piston rings are used. The top ring is a chrome-faced, rectangular-section ring. The second and third rings are tapered-faced compression rings. An oil ring located above the piston pin and one located

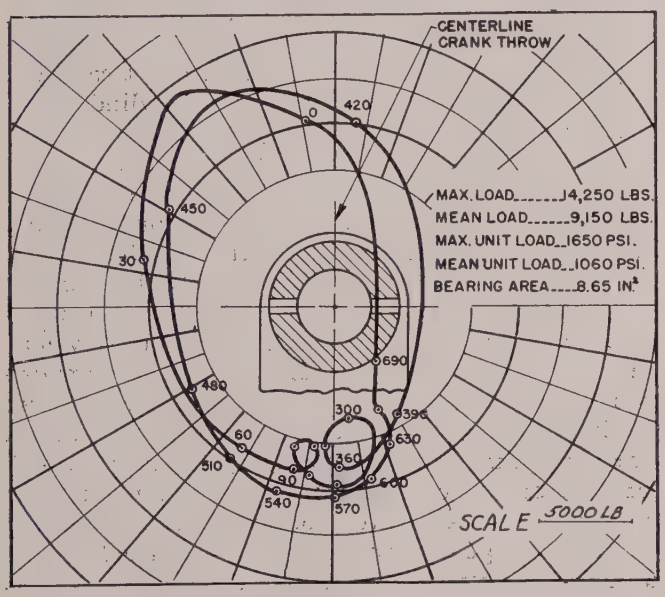


Fig. 7 – Crankpin journal loads with respect to crank throw, for 12-cyl, 600-hp engine at 2000 rpm (numbers on diagram refer to crank angles, 0 deg with reference to No. 1 left-bank piston at tdc on power stroke)

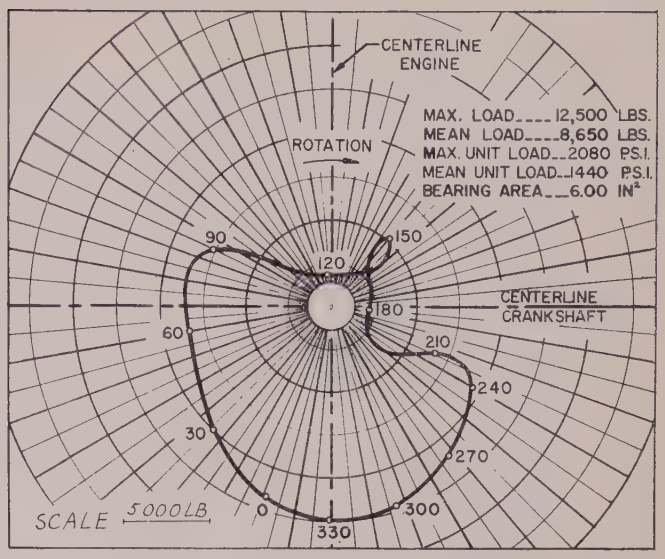


Fig. 8 – Center main bearing loads, for 12-cyl, 600-hp engine at 2000 rpm (numbers on diagram refer to crank angles, 0 deg with reference to No. 1 left-bank piston at tdc on power stroke)

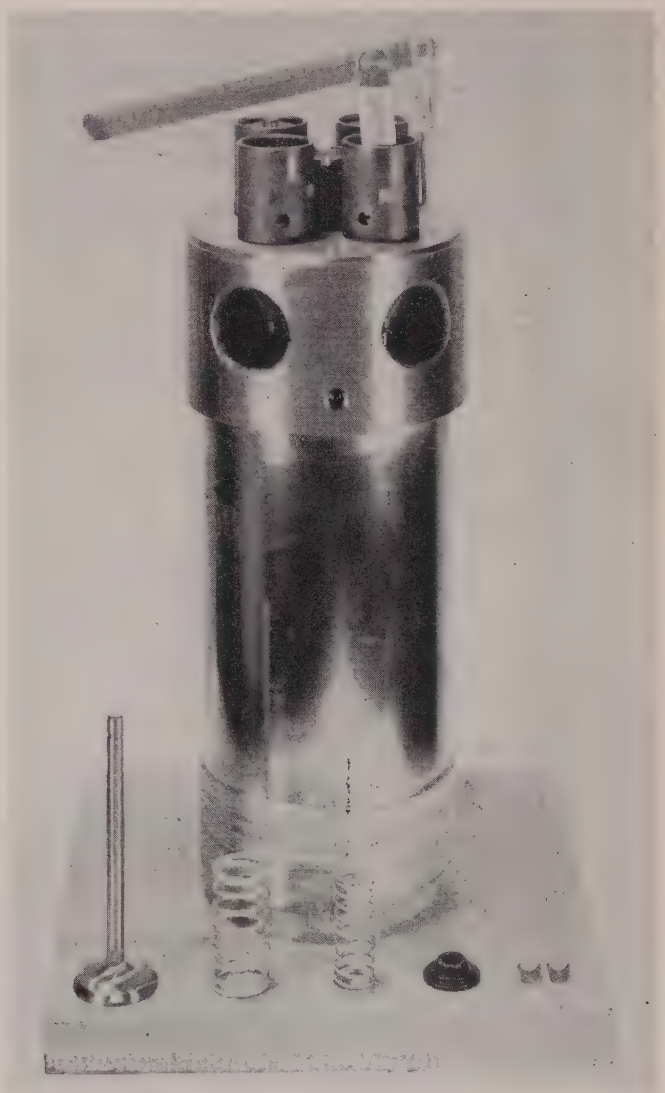


Fig. 9 – Unit cylinder and head assembly

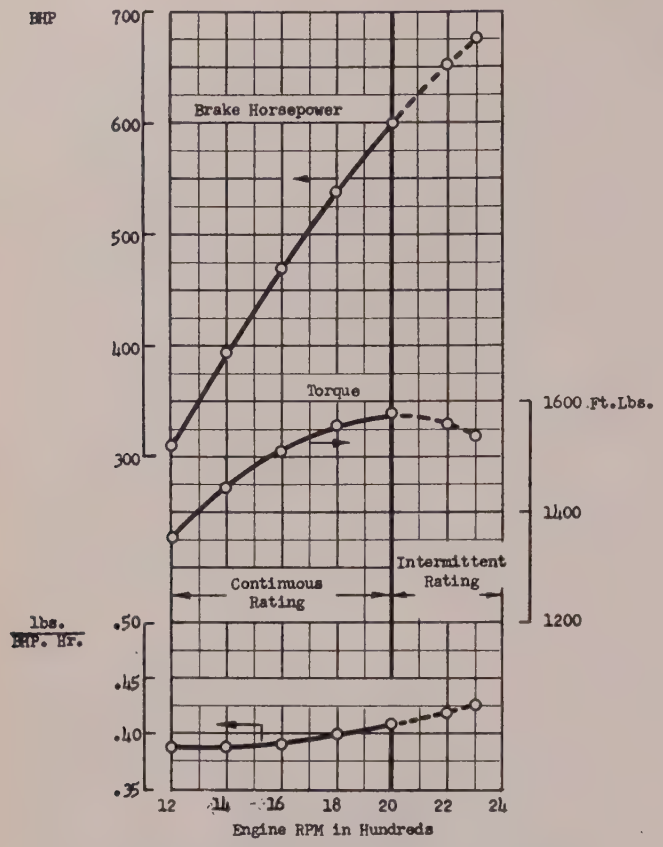


Fig. 10 – Continuous power rating of 12-cyl engine

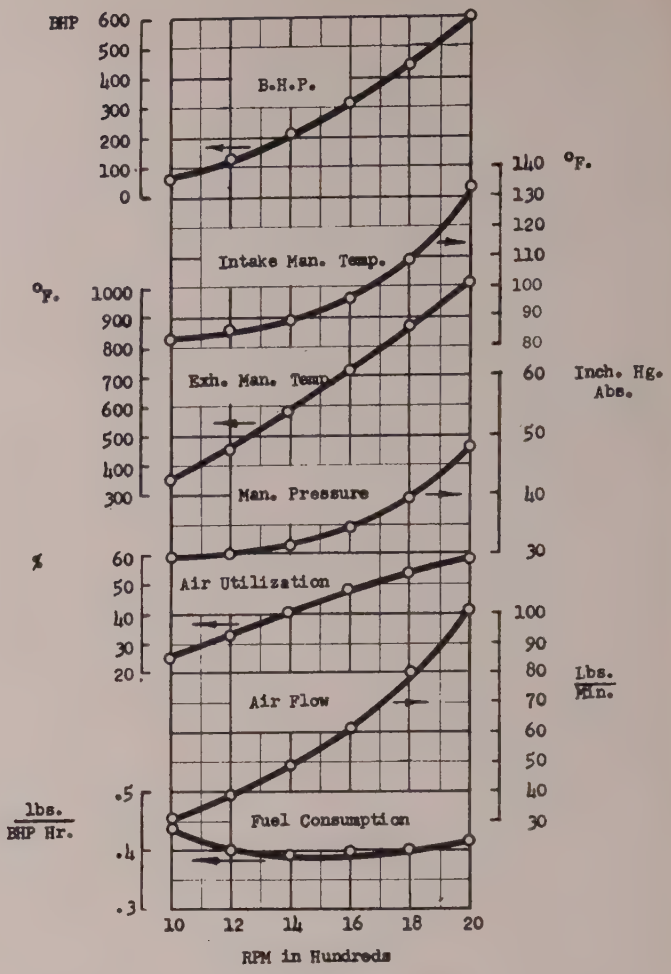


Fig. 11 – Performance corresponding to assumed propeller load, for 12-cyl engine

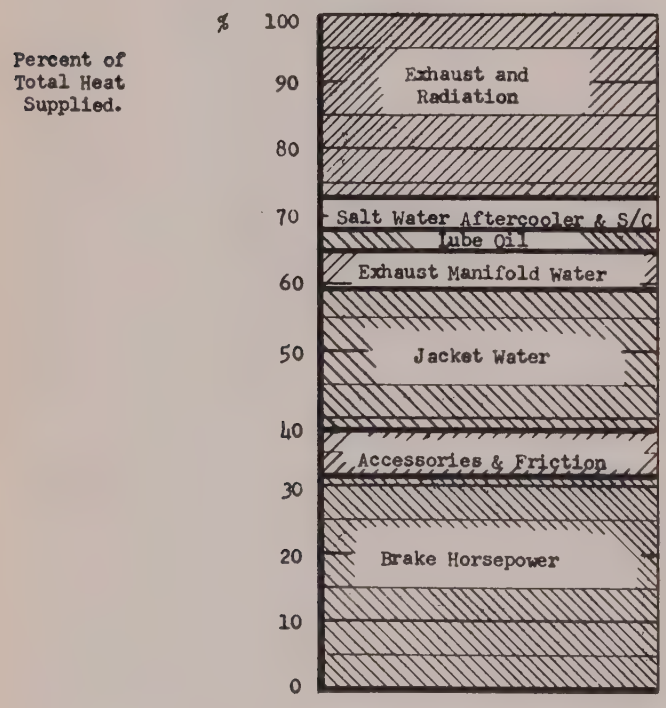


Fig. 12 – Heat balance for 12-cyl engine at 600 bhp and 2000 rpm, based on 19,000 Btu per lb fuel

at the bottom of the piston skirt are identical and incorporate liberal slotted oil passages with tapered-faced lands. The piston pins are full floating and aluminum buttons are located at each end of the piston pin to provide a proper cylinder contacting surface.

All accessories are driven by gears located at the flywheel end of the engine. Most of these gears are mounted on antifriction bearings for reliability in operation and starting. With the drive from this end of the engine, gear life is extended and gear noise minimized. Most accessories are separately removable and are located where they may be reached easily. Provisions are made for driving two 6-cyl fuel pumps (one for each bank), two fresh-water pumps (one for each bank), an oil pressure pump, an oil scavenging pump, an auxiliary oil pump for use with connected machinery, a governor, salt-water pump, starting motor, tachometer generator, and an overspeed cutout actuating governor. The latter consists of a spring-loaded rotating element, which vents an oil line connected to the valve actuating plunger so that low oil pressure will be maintained in the connec-

tion during all operation. In case the engine speed is increased above the setting of the overspeed governor, the vent is closed off by movement of the spring rotor, resulting in an immediate sharp rise in oil pressure at the far end of the connecting line, operating the plunger at that end to release a valve closing off the air supply to the engine.

The exhaust gases leaving the cylinder enter a water-cooled exhaust manifold, continuing into the gas turbine side of the supercharger. The exhaust manifolds and the supercharger turbine are water-jacketed in order to eliminate hot surfaces, which are objectionable in marine applications both as a matter of comfort and safety. Insulated stainless-steel tubes are inserted in the exhaust manifolds to reduce the heat loss through the jackets and thus help turbine performance.

The air leaving the supercharger is cooled by a heat exchanger, thus contributing to minimum fuel consumption.

Multiengine Performance

The continuous power rating of the 12-cyl engine is shown in Fig. 11. This also shows intake air slightly more than 300 is developed at 1200 rpm. The torque curve rises as speed increases, showing 1580 ft-lb at 2000 rpm. The specific fuel consumption is slightly over 0.4 lb per bhp at 2000 rpm. These are without the benefit of correction factors. Since the original service applications of these engines involve marine propulsion uses, performance corresponding to an assumed propeller load is shown in Fig. 10, with 600 hp at 2000 rpm; temperature, exhaust gas temperature, intake manifold pressure, airflow, and air utilization.

Fig. 12 shows heat balance, based on the high heating value of the fuel.

More detailed performance information on temperatures, pressures, airflow, and the like, corresponding to supercharged engine operation, is given in Fig. 13.

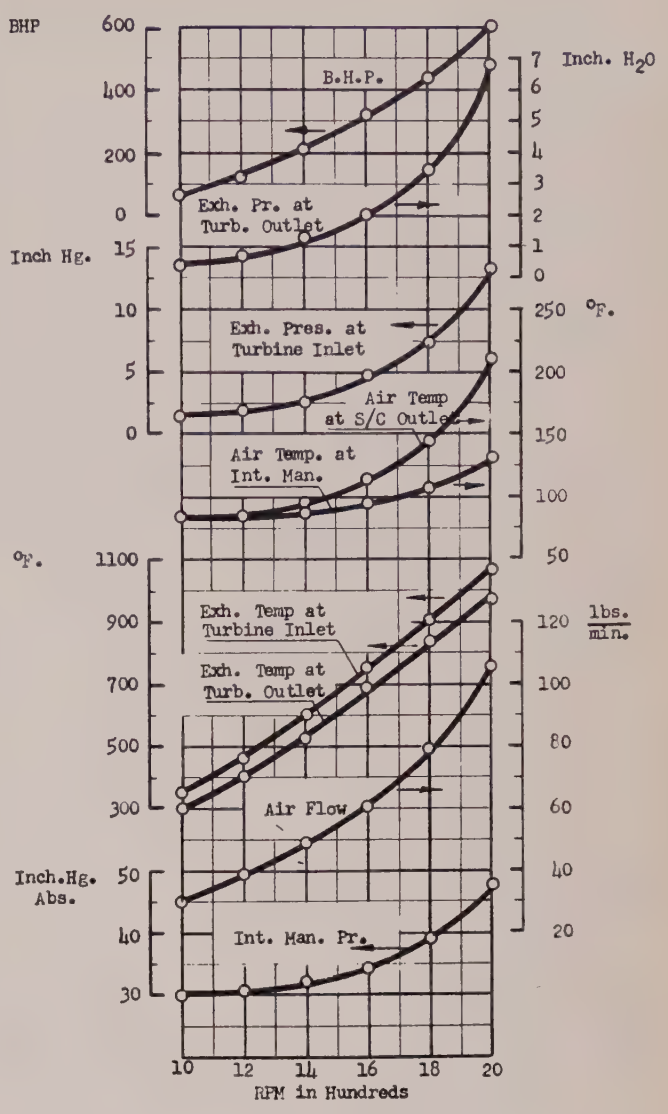


Fig. 13 – Supercharged-engine data – barometer 29.6 in. of mercury, air inlet temperature 88 F

Injection System Performance

The fuel injection pumps are conventional American Bosch units, except that the main housings are made of aluminum and the plungers have special features related to fuel timing control. The nozzle is of conventional pintle design. The pump plungers are 10-mm diameter; the pintles in the nozzles are 2.0-mm diameter.

The pump plungers have helices for both the start and finish of injection, as shown by the pump diagram in Fig. 14. These have been specified to provide timing more suitable than could be obtained with a single helix. This has helped to compensate for the omission of a variable timing device.

A land is formed in the working zone of the plunger occupying the first 0.2 in. of rack travel, to provide a fixed port opening and port closing for this portion of the rack. This results in a special timing and quantity of fuel particularly favorable for starting. This land extends over a sufficient portion of the rack travel to permit the special starting setting to be obtained without precise positioning of the rack. One application of the engines required positive starting in minimum time and this starting land was a useful aid in meeting this objective. Fig. 15 shows the prompt starting that is obtained and rapid acceleration at normal room temperatures.

The rack is ordinarily connected to a hydraulic governor. A control arrangement is provided by means of which the governor connection is modified to establish the starting land position of the rack while starting. The control is then moved beyond the run position to give sufficient rack movement until the governor becomes operative. In the run position the governor has full control. Zero fuel delivery, as required for stopping, is

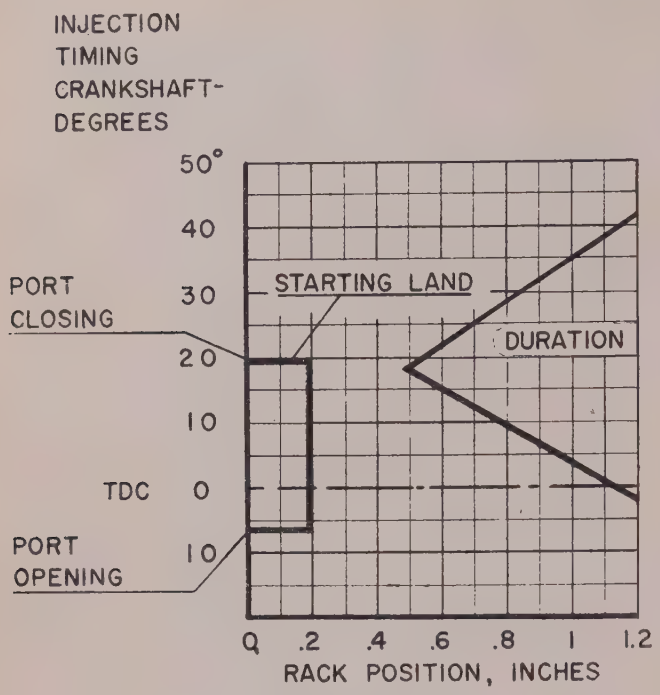


Fig. 14 – Static timing curve

obtained by moving the rack to a position between the starting land and double helix portions.

The performance data that have been presented through Fig. 15 apply to the engines as they are being built now. A series of survey tests was made in the early development stages to determine the effect of several variables, and the results of some of these tests will be presented. In most of these survey tests, the pump and nozzle combinations used differed from the production specification, so that the data are not directly comparable. The early tests were most important in establishing the production specification and at this time serve to give a better understanding of the behavior of the combustion system.

The relation between the beginning and the end of pump action as determined from static measurements, and the beginning and the end of injection into the cylinder and the start of actual combustion, as measured with the engine operating, is shown in Fig. 16. These data apply to a lower helix 12-mm plunger.

The beginning and end of injection were determined with electronic equipment utilizing a linear variable differential transformer providing a variation in output dependent on the position of a small armature fastened to the needle of the injector. The type of record produced by this equipment is shown in Fig. 17.

Fig. 18 shows the relation between the injection pump action, the injection nozzle needle lift, and the pressures developed in the cylinder at rated load and speed. Measurements were also made of the pressure in the precombustion chamber. While these diagrams indicate that pump action has been completed before the needle lift begins, the action relation is not known since the pump delivery functions are static measurements. The diagrams indicate rather rapid lift of the needle, with greater displacement at the start than at the end.

Effect of Compression Ratio Changes

Single-cylinder tests were made with four different compression ratios 14, 15, 15.5, and 16/1. Data

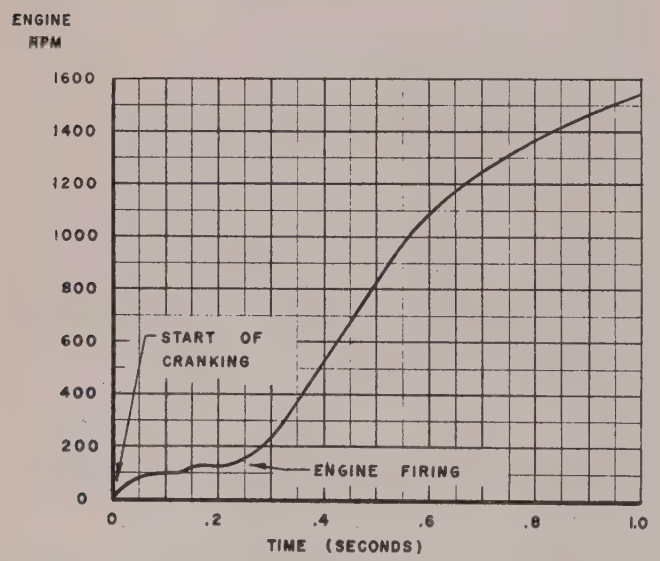


Fig. 15 – 12-cyl engine starting test – room temperature, free engine, starting characteristics as measured by oscillograph, using starting land on injection pumps

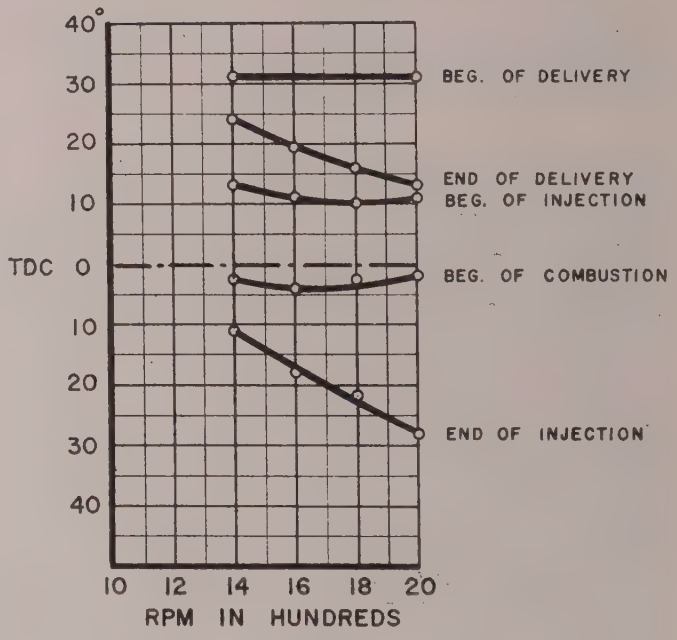


Fig. 16 – Single-cylinder engine – needle lift was measured electrically by using linear variable differential transformer

showing brake horsepower, maximum pressures, and specific fuel consumption as a function of engine rpm for the four compression ratios are given in Fig. 19. These tests show conventional effects. It is of interest to note that there is less variation in the specific fuel consumption over the speed range at the highest compression ratio. These tests were made with a constant fuel quantity.

The curves of Fig. 20, showing the result of vary-injection timing, indicate no important change in sensitivity of timing due to changes in compression ratio.

Supercharging

The effects of varying supercharger pressure were also investigated on the single-cylinder engine. Data obtained at 14/1 compression ratio and with variations in absolute intake manifold pressure from 50 in. of mercury to 70 in. are presented in Fig. 21. These curves were obtained with a fixed exhaust temperature of 1200 F. These curves show that the compression pressure as estimated for the diagrams varies in direct proportion to the absolute manifold pressure, while the maximum, or firing pressure, increases at a slower rate. This situation is useful in applying the cylinder for higher output. Substantial increases in power can be obtained without encountering extremely high pressures.

The favorable pressure relations shown by these curves are further substantiated by pressure-time diagrams taken with an oscilloscopic indicator.

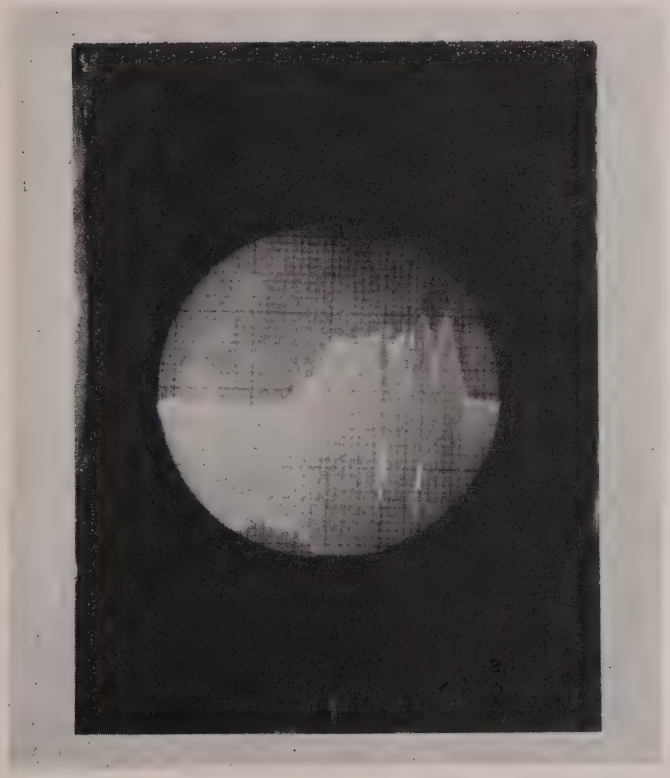


Fig. 17 – Injection nozzle needle lift diagram, measured with linear variable differential transformer

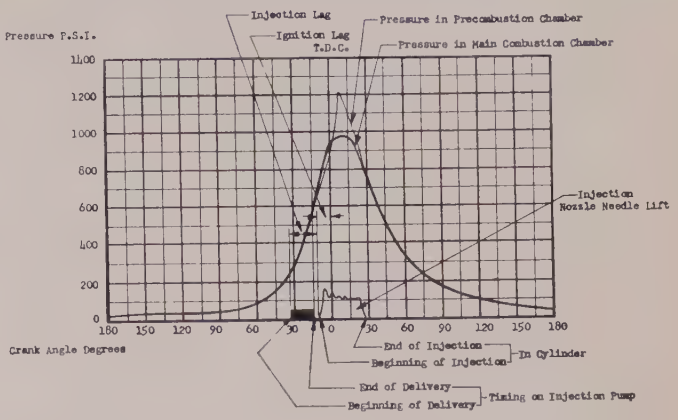


Fig. 18 – 12-cyl engine pressure-time diagram and needle lift diagram – 600 bhp, 2000 rpm, 140 psi bmep, 16/1 compression ratio, 46 in. of mercury absolute manifold pressure

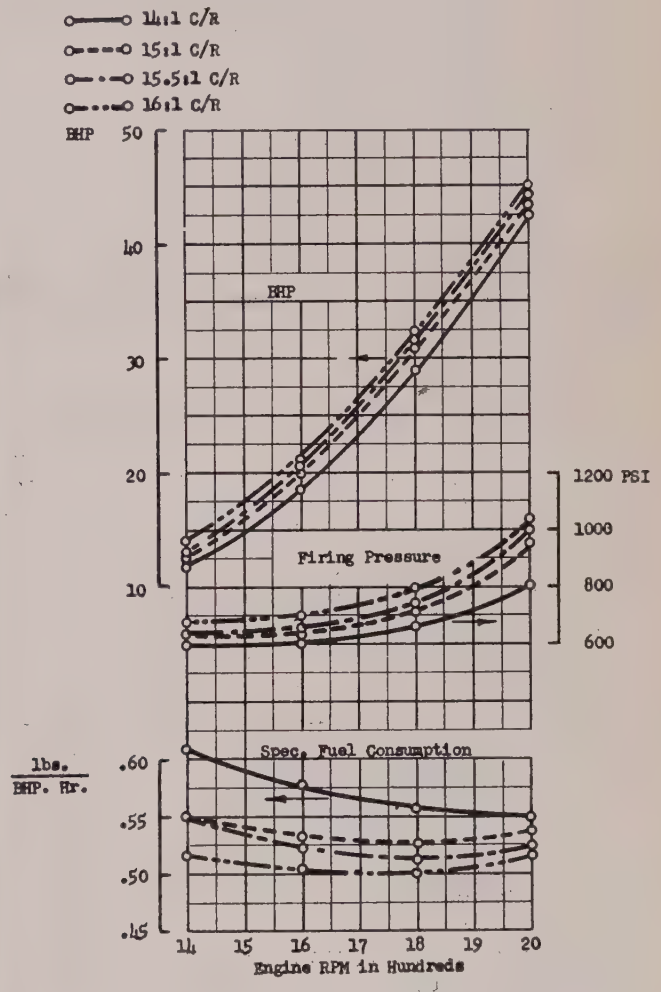


Fig. 19 – Effect of varying compression ratio, single-cylinder engine – tests run at constant fuel rate

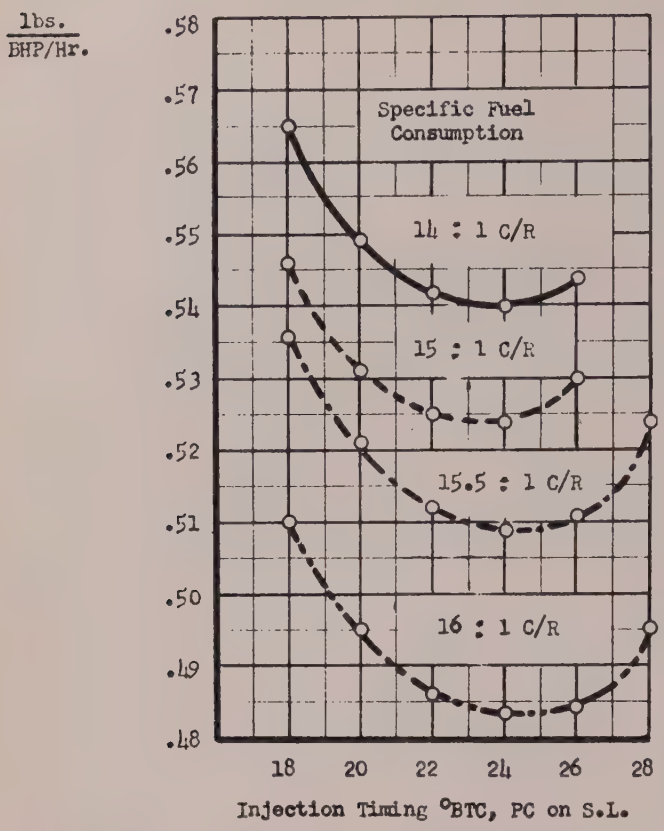


Fig. 20 – Fuel injection timing curves for single-cylinder engine, 2000 rpm, 42.7 hp

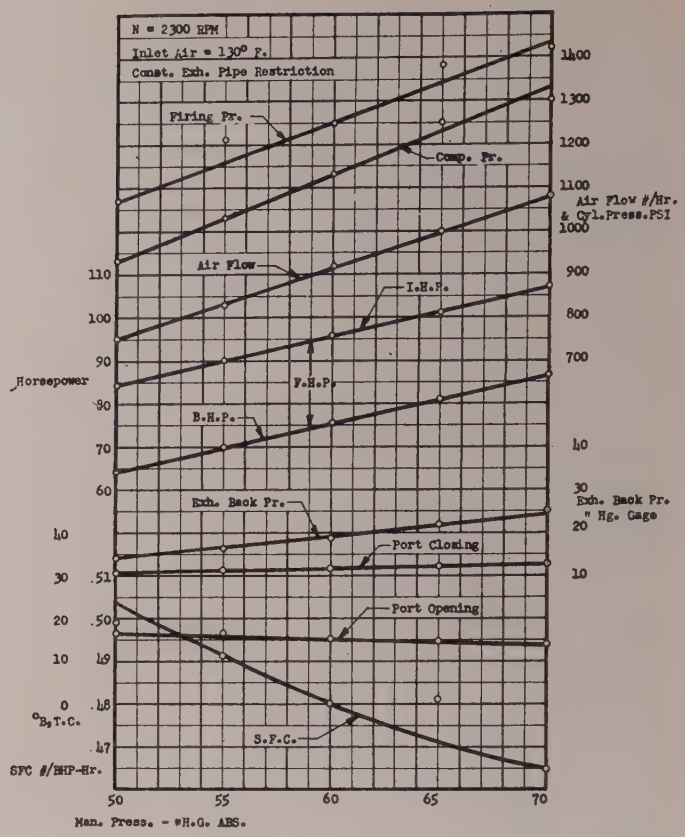


Fig. 21 – Effect of varying absolute intake manifold pressure – single-cylinder engine

Oscilloscopic records are shown in Fig. 22. These were taken at an absolute manifold pressure of 48 in. of mercury. Fig. 23 shows the effect of changes in injection timing.

The curve with 21.7-deg port closing approaches the ideal diesel diagram. As the timing is advanced to 26.2 deg, the pressure rise becomes faster and the amount of rise, greater. The diagrams are obviously quite favorable for extended service life as compared with the distinctly sharp rise that is obtained with many direct-injection engines.

The present multicylinder engines have a continuous rating of 50 hp per cyl. It is noted that this same cylinder in these single-cylinder tests shows over 85 bhp at 70 in. of mercury. The friction relations of the single-cylinder and multicylinder engines are such that still greater horsepower per cylinder would be developed under the same conditions on the multicylinder engine.

The first multicylinder engine of this type to be built was equipped with a mechanical supercharger, and in preparing for the present series of engines, comparative tests were made with a mechanically driven centrifugal supercharger and a turbosupercharger. Some results of these tests are shown in Fig. 24. At the 300-hp continuous rating of the 6-cyl engine:

1. The exhaust temperature is 100 deg higher with the turbosupercharger,
2. The specific consumption is about 5% lower.

It will be noted that the airflow is considerably less with the turbocharger, which may account partially for the higher exhaust temperatures. These results were obtained with 90-deg valve overlap and normal exhaust opening and intake closing functions.

Nonmagnetic Construction

A special and unusual requirement experienced in some applications involved high nonmagnetic content. These engines as basically designed were attractive for this purpose, since aluminum was used in considerable quantity, where cast iron or steel is the more usual material. The nonmagnetic content was increased by using bronze and stainless steel in place of steel, where aluminum was not adequate. Further increases in nonmagnetic content were accomplished by the use of nitrided stainless steel and nonmagnetic nickel alloys in highly stressed parts, such as crankshafts and connecting rods. In this way, the nonmagnetic content was raised to 80% in one engine.

Since there was no suitable information on the fatigue characteristics of nitrided stainless steel, a series of fatigue tests was made. These tests employed an unusual test specimen form including a

Fig. 22 – Cylinder pressure diagrams, injection timing, 20 deg btc port closing on starting land, for 12-cyl diesel – peak pressure 1080 psi, compression pressure 1020 psi

notch effect more nearly representative of engine parts than the standard specimen. The notch tends to lower values. Some values of interest are:

X4340-302 Brinell	28,000
VCM-Rockwell C-30	32,000
VCM-Nitrided	67,500
347 Stainless	22,000
347 Stainless-Nitrided	38,500

As in other tests, nitriding raises the endurance limit considerably. While the endurance limit of the nitrided stainless is less than nitrided VCM, it is still greater than a 40-carbon, strong alloy steel.

Applications

In concluding this presentation, a series of illustrations will show some applications of the engine.

Fig. 25 shows the 6-cyl, 300-hp engine combined with a 3.2/1 reverse and reduction gear. Fig. 26 is a sectional view of the reverse gear. The output shaft is coaxial with the crankshaft. The forward and reverse clutches are hydraulically operated and the oil supply required for this purpose and for lubrication of the gear is separated from the engine oil in order to avoid contamination by any dirt-making conditions that may arise in the engine. No separate reversing gears are provided. Each clutch has a separate sun gear, which meshes with a single

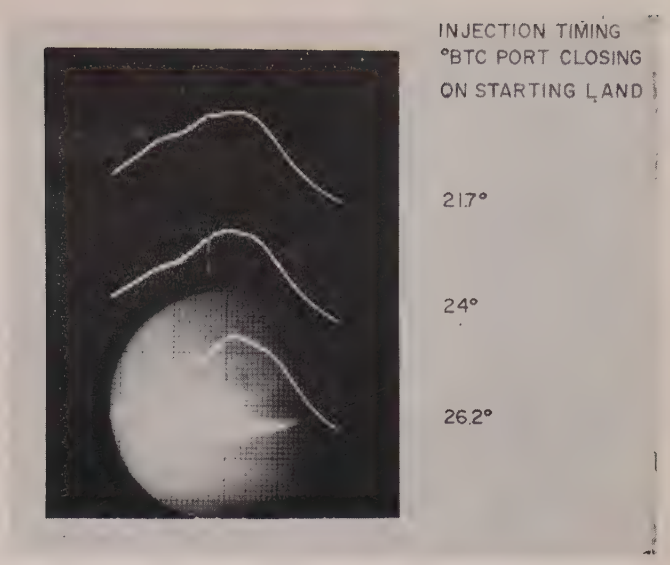


Fig. 23 – Cylinder pressure diagrams for single-cylinder engine

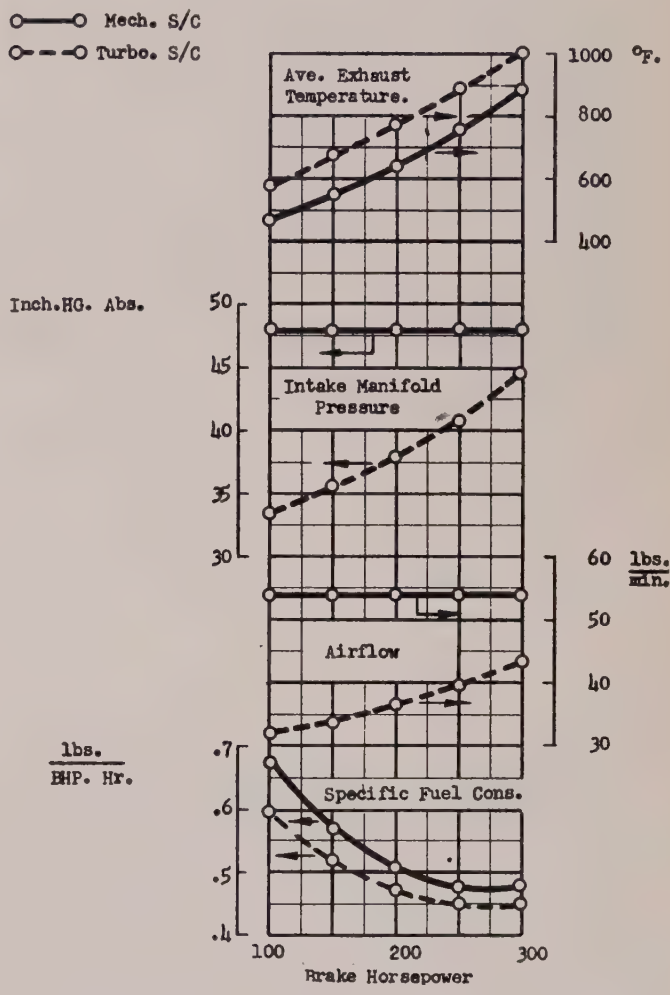


Fig. 24 – Mechanical supercharger performance compared with turbo-supercharger performance – 6-cyl engine, 2000 rpm

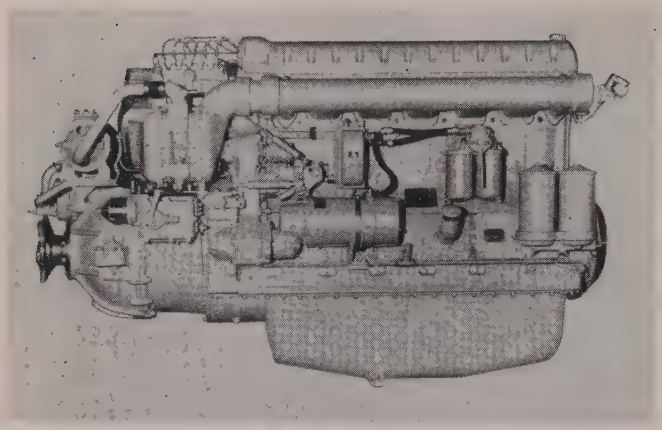


Fig. 25 – 6-cyl, 300-hp engine combined with 3.2/1 reverse and reduction gear

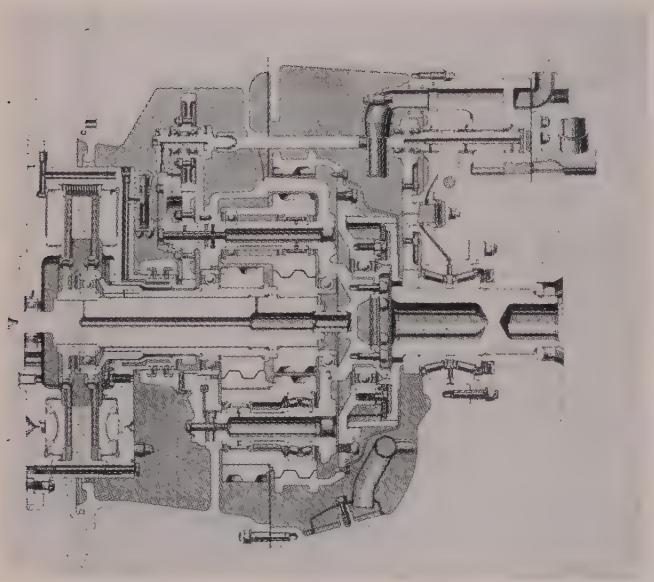


Fig. 26 – Sectional view of reverse gear

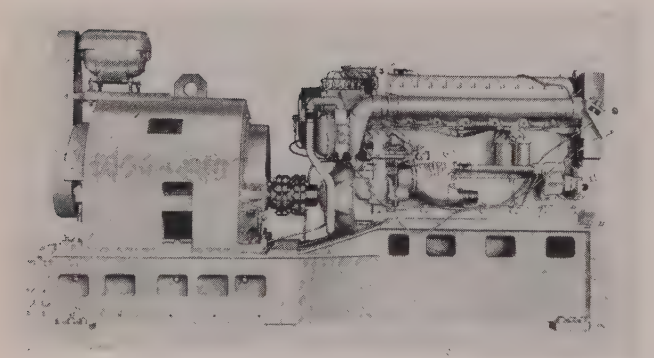


Fig. 27 – 6-cyl, 300-hp engine arranged for driving general-purpose a-c generator

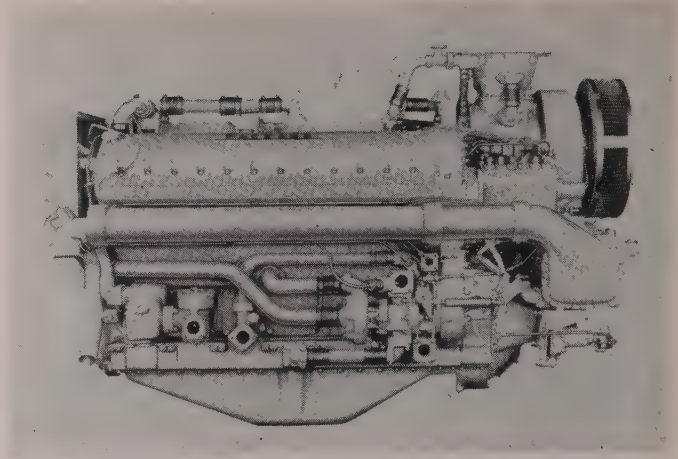


Fig. 28 – 12-cyl, 600-hp engine arranged for connection to propulsion gearing

ring gear through two planet systems—one for each direction of rotation.

Fig. 27 shows this same basic engine arranged for driving a general-purpose a-c generator.

Fig. 28 shows the 12-cyl, 600-hp engine arranged for connection to propulsion gearing.

Fig. 29 shows the same basic engine modified by the adaptation of a 5/1 reverse and reduction gear. This reverse gear is also hydraulically operated. In this case, the output shaft is offset from the crankshaft. This gear embodies several unique features. Of special interest is the arrangement for transmitting the propeller thrust through the reduction gear housing to a thrust bearing located on the forward side of the housing. The thrust bearing is mounted in a carrier, which is guided by the housing but permits the carrier to move longitudinally with relation to the housing. The carrier is restrained by a strut, which connects to the engine mounting structure, thus relieving the entire engine structure of thrust loads. The "bull gear" is mounted on separate bearings arranged to take the main gear reactions, and driving torque to the shaft is transmitted by a thin web. This is particularly favorable for lightweight construction.

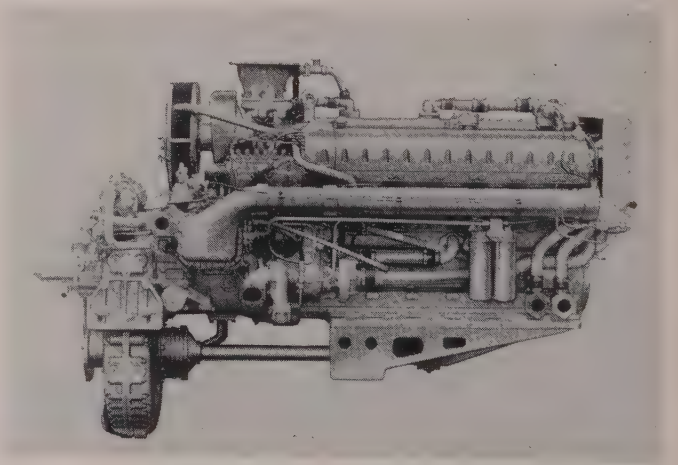


Fig. 29 – 12-cyl, 600-hp engine modified by adaptation of 5/1 reverse and reduction gear

DISCUSSION

Likes One-Piece Cylinder Construction

– A. W. Pope, Jr.
Waukesha Motor Co.

NORMAL peak combustion pressures are getting so high both in spark-ignition and supercharged diesel engines that design improvements must be made in the cylinder-head joint. The new Packard diesel engine contains an ingenious solution to this design problem. I refer to the one-piece cylinder and head construction.

This development by Packard with a blending of European and American design motifs seems to be somewhat analogous to the great American biological melting pot which has produced such outstanding results in the social structure of the world. It is easy to recognize Daimler-Benz influence in the one-piece cylinder design. The famous high-output German E boat engine and other Daimler-Benz race and aviation engines have all been built around a one-piece cylinder construction, but American techniques and mental climate may have been necessary to create this new design.

Aside from the high manufacturing cost of one-piece cylinder construction, inaccessibility has always been the main objection. This may no longer be a valid criticism now that we have valves, piston rings, and cylinder bores that will operate well over 100,000 truck miles without maintenance. It may now be possible to design an engine not requiring frequent top overhaul. If this be the case, the one-piece cylinder design becomes very desirable, with its elimination of gasket troubles, cylinder and valve-seat distortion from stud loads, and high weight.

The reason for our enthusiasm for one-piece cylinder construction is our experience with the CFR-ASTM gasoline knock test engine cylinder. We have been manufacturing this one-piece cast cylinder for 23 years, with complete freedom from gasket sealing and cylinder-head stud distortion problems.

The standard ASTM gasoline knock test methods are run with the engine operating continuously under heavy detonation with 212 F jacket temperature. Some laboratories operate them on a 24-hr day for months at a time. The one-piece cylinder construction seems to be unaffected by continuous operation under heavy detonation.

The extremely efficient cooling obtainable with one-piece cylinder construction is another feature sometimes overlooked. Elimination of stud bosses and cylinder-head joint flanges in the water jacket improve the water circulation and cooling efficiency to a very unusual degree. To illustrate, the one-piece CFR cylinder operates with thermosyphon circulation without a water pump at 300- to 400-psi imep with stable cooling conditions. Even with glycol coolant at 370 F temperature, there is no tendency toward steam pocketing or unstable cooling conditions. Such cooling efficiency is not obtainable with the obstructed water jackets that are necessary with a removable cylinder head.

ORAL DISCUSSION

Reported by Vincent Ayres
Eaton Mfg. Co.

J. W. Pennington, Koppers Co., Inc.: Has piston cooling been investigated and found necessary? What is the effect of continuous operation on the oil?

Mr. Ware: Piston cooling was provided in the original design, but was not required when operating at 140-psi bmep. If, however, later on, higher outputs indicate a need for it, the design is still available and could be used without any major change. Piston temperature with the present design seems well controlled because of the excellent cylinder cooling of the design. 100-hr tests produced no difficulty as far as the oil was concerned. We intend to make more tests to investigate this matter further. Specification No. 30 oil was used satisfactorily.

Russell Pyles, Van Der Horst Corp. of America: I would like to discuss Fig. 6, particularly in regard to the use of horizontal tie-rods for stiffness. Engine roughness might be present with a weak construction because of the forces set up from the cylinder main bearing loads. The load diagram has horizontal components that, in the Packard design, were resisted by the tie-rods, which nearly doubled the stiffness, and probably account for the smooth operation.

Cylinder wear can be classified under three main headings:

1. A normal amount of wear would be low under constant running with a good fuel.
2. An abrasive type of wear, which could be traced to dirt and ash deposits from combustion.
3. Corrosive wear, which is particularly troublesome with low temperatures, high sulfur in the fuel, and long idle periods. I would like to differentiate between idle and idling. The idle period referred to means long periods of time without engine operation.

In military usage difficulty had been experienced where periods of prolonged standing created rusting and pitting of cylinder bores. This problem was completely overcome by chrome plating the cylinder walls. If the Packard engine is used for marine operation in naval service, a similar problem could arise. Commercial operation with poor fuels that are sometime used could create a cylinder wear problem, so I would suggest consideration of chrome plating as a means of improving durability.

Mr. Ware: I agree that the tie bolts are particularly useful with lightweight aluminum engines, to increase resistance to transverse loading. I also recognize the possibility of corrosive cylinder wear, but to date we have experienced no difficulty in this regard. We feel that this may be due to the nature of our design.

O. D. Treiber, Hercules Motors Corp.: Do you have any trouble caused by deposit formation around the cylinder head, which might freeze it to the block and prevent removal without damage? Sometimes fuels, loads, and temperatures are involved in the creation of these deposits.

Mr. Ware: We have not experienced this trouble with any of the several variations of design that we have tested. If such a condition ever arises, removing the large cylinder holding nuts and then cranking the engine over would create considerable pressure in the cylinder, due to compression, which would no doubt force the cylinder past the deposits without any harmful distortion.

Authors' Closure To Discussion

IN connection with Mr. Pope's comment on one-piece cylinder background, this feature has been a prominent part in almost all Packard-designed high-output engines. It goes back to aircraft engines built before World War I. It was used again in the Liberty engine, which was developed during World War I. Later, in the 20's, it was used in the models 1500 and 2500 water-cooled, gasoline aircraft engines and in the aircraft diesel. Then, more recently, it was used in the larger gasoline engines that powered PT and aircraft rescue boats. This extended experience developed the merits of this type of construction for high-output performance and light weight, which were considered well worth while for the new diesel engines.

Service and Testing Observations of

HAVING utilized the past decade to develop and put into service a number of production helicopters, the industry now finds itself faced with the task of improving the service life of their machines to a degree approaching that of the fixed-wing aircraft. Outstanding among the deterrents to satisfactory service life, and perhaps most peculiar to the helicopter industry, is the phenomenon of progressive or fatigue failure. This particular type of failure has, for the past century, been the object of considerable scientific endeavor, speculation, and controversy. To date, the basic facts that allow the designer to consider the fatigue phenomenon are primarily empirical in nature and have been accumulated through a century of basic experimentation, specific engineering tests, and actual in-service observations.

Fig. 1 – Hiller 360 series UH-12B model helicopter in flight

It is the contention of the authors that interchange and dissemination of information in the latter two categories is somewhat relaxed in the helicopter industry. As such, the purpose of this paper is to present some specific results and observations that fall into these latter two categories.

The subject matter for this paper, for the most part, is a direct result of four years of operational experience with the Hiller 360 series helicopters (see Fig. 1). The A model helicopters were put into service in 1949, and the B models, which incorporated a power increase from 178 hp to 200 hp, were put into service in 1951. Commercially, the helicopter has been subjected to a variety of uses and environments, while the military version has been utilized primarily for training purposes. One-hundred and ninety-six A models were manufactured and to date they have accumulated a total of 35,000 flight hr, the highest for an individual aircraft being 1200 hr. Three-hundred and five B models have been placed in service. They have accumulated 23,000 flight hr, the highest individual time being 330 hr.

Fatigue Failure Incidence

During the 58,000 hr of flight time that have been accumulated on these aircraft, a total of 169 failures that can be attributed to fatigue have been reported. The fatigue failure incidence by component groups is shown in Fig. 2. As may be seen, the failures in basic structure exhibited the lowest percentage. The transmission, drive system, and those

Fatigue Failure in Helicopter Components

Richard M. Carlson and F. David Schnebly
Hiller Helicopters, Inc.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

parts that are classified as miscellaneous brackets exhibited the highest number of fatigue difficulties. Considering the component percentages shown in Fig. 2, they are, in general, what might be expected, since they seem to be greater in the parts or systems where the loads are high and where frequent or continuous load reversals take place. This reasoning, however, is not sufficient to account for the high 32% figure shown under miscellaneous brackets. Probably the highest offenders in the bracket classification are parts like the sheet-metal oil cooler attachment brackets and engine induction system mounting brackets. The reason for their high failure incidence might be explained in that they represent components that receive the least amount of consideration in the design phases. This lack of attention could be attributed to the erroneous concept that such brackets are lightly loaded, and that their failure is not necessarily disastrous to the aircraft.

Fatigue Failure Incidence Origin

All of the fatigue failures have been classified into one of the three groups shown in Fig. 3, depending upon whether their origin could be traced to design error, manufacturing error, or field error.

FATIGUE failure is now one of the most serious hindrances to satisfactory service life for helicopters.

To help the industry bring the fatigue failure problem under control, these authors summarize what they have learned from studying 169 fatigue failures that occurred during 58,000 hr of helicopter flying operations.

They have classified these failures as being caused by:

1. **Design error.**
2. **Manufacturing error.**
3. **Field error.**

They discuss specific examples in each group and what was done to correct the cause of the failure.

The Authors

RICHARD M. CARLSON is chief structures engineer for Hiller Helicopters. He obtained both B.S. and M.S. degrees in aeronautical engineering from the University of Washington. Mr. Carlson has been with Hiller since 1947.

F. DAVID SCHNEBLY is a structures engineer for Hiller Helicopters. He received his B.S. degree in aeronautical engineering from the University of Southern California. Before joining Hiller, Mr. Schnebly was a research assistant at the university.

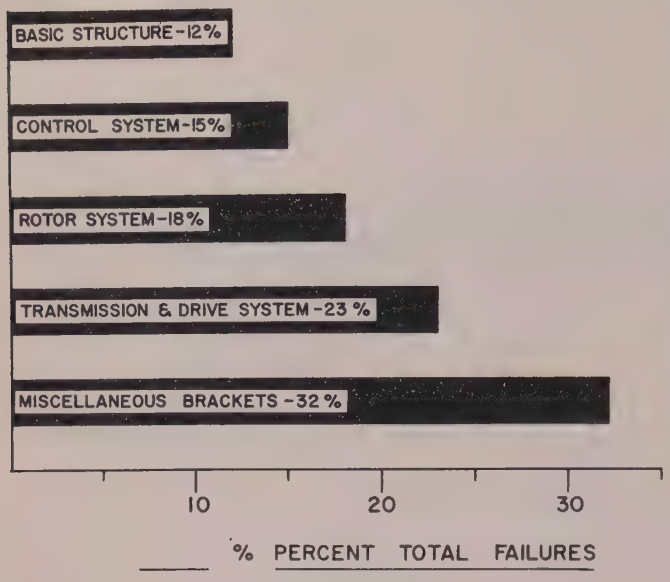


Fig. 2 – Incidence of fatigue failures in component groups – based on UH-12A and 12B model helicopters with total accumulative service of 58,000 hr on 322 aircraft

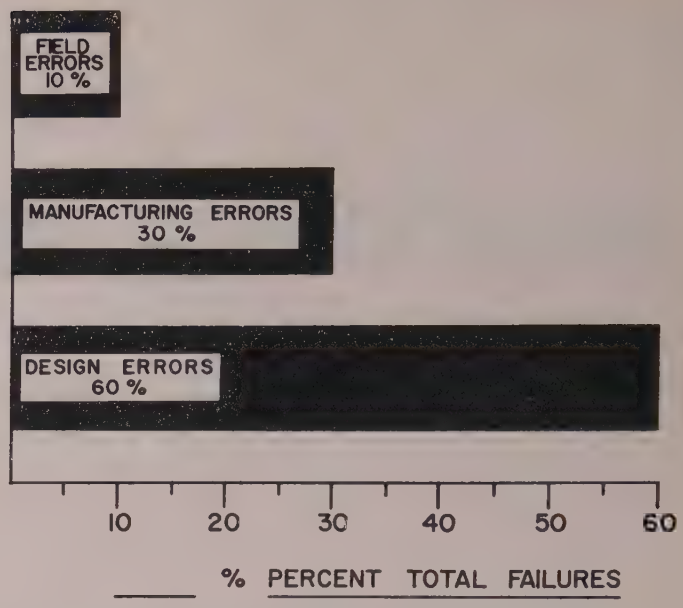


Fig. 3 – Incidence of fatigue failure origin – based on UH-12A and 12B model helicopters with total accumulative service of 58,000 hr on 322 aircraft

The failures, in general, fit very nicely into one of these three classifications. Several specific failures will be presented as examples of these three groups.

Design Errors—Failures having their origin in the design stages will be discussed first. Fig. 4 shows the SAE 4130 steel collective pitch incidence arm used on the 360 series helicopter for applying collective pitch to the main rotor blades. The arrows point out the cracks in a fillet radius that brought about the fatigue failure of the part. This part receives both steady and oscillatory loads in flight, which are the aerodynamic reactions to the pilot-applied loads on the collective control stick. After very thorough study and examination, it was found that the cracks were the result of hydrogen embrittlement in the steel; the hydrogen embrittlement being brought about by the chrome plating of the part. Chrome plating has been discontinued and a cadmium plate and baking process is now used in its place with no additional failures having been reported.

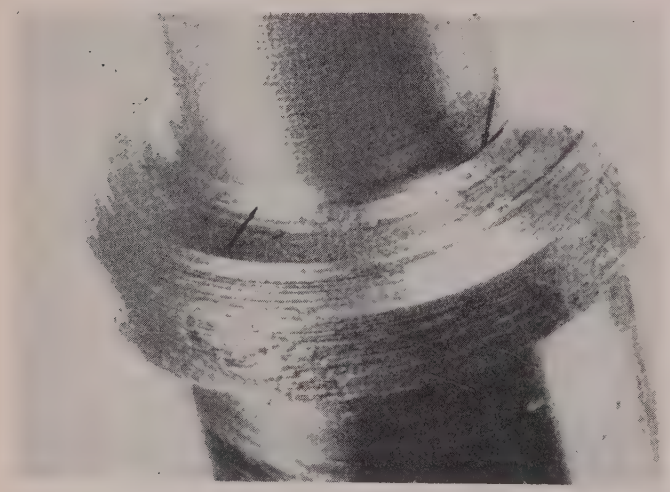


Fig. 4 – Photo showing start of fatigue failure in SAE 4130 steel collective pitch incidence arm

Another failure that was due to detail design is shown in Fig. 5, which depicts a failed tail-rotor-drive slip-joint centering pin. The centering pin, which failed in bending at the section where it enters the slip-joint yoke, is shown above the yoke, with the remaining portion still left in the yoke below. This pin is pressed into the yoke on the tail-rotor driveshaft and is used to locate and center the cam follower tee in the cam of the slip-joint yoke itself. In the original design the intent was that the pin would act as a centering device only, and therefore would have little or no bending moment imposed upon it. However, the tolerance buildup and misalignment problems of the entire tail-rotor-drive installation were not properly considered, and high oscillatory bending stresses were found to be present in the pin due to these causes.

This condition was relieved by shortening the pins and opening the tolerances so that the possible angular misalignment in the system could be taken by the joint without producing a bending moment in the pin. Again, this failure would be classified in the detail design error category.

Another factor leading to failures due to design inadequacies is the lack of knowledge concerning the load spectrum. An excellent example of this is the failure shown in Fig. 6. This figure shows

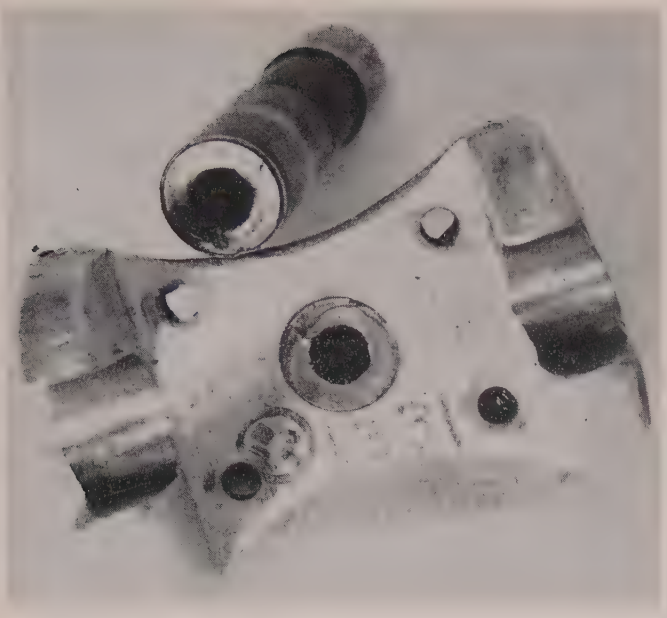


Fig. 5 – Fatigue failure of tail-rotor-driveshaft slip-joint centering pin

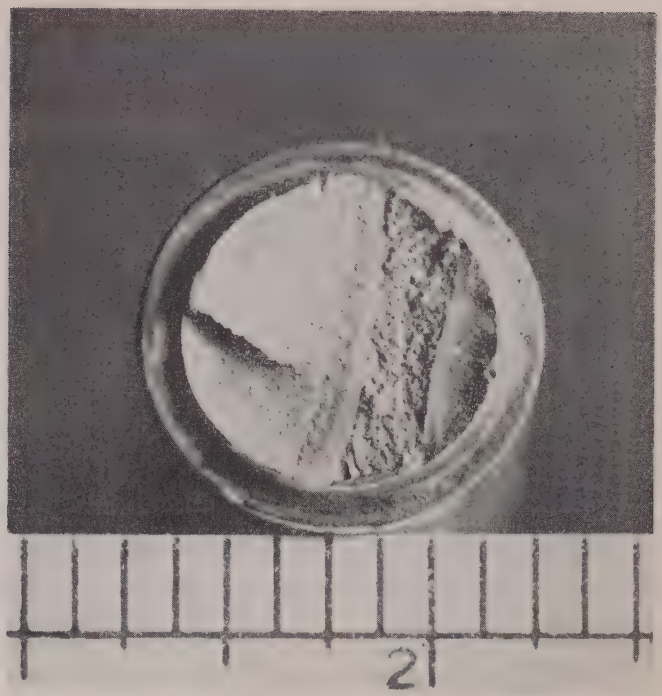


Fig. 6 – Fatigue failure of tail-rotor pitch-change control rod

the fatigue failure of a tail-rotor pitch-change push-pull rod. The original design of the part was based on calculations that, at the time, seemed quite realistic. The small loads found gave rise to the selection of SAE 1095 steel drill rod. These loads were determined, however, without much thought being given to any possible resonance phenomenon or chordwise unbalance effect. Bending loads, determined by flight tests, were found to be considerably higher than those calculated, and occurred in an alternating fashion at a frequency of one and three per revolution of the tail rotor. The one-per-revolution bending was attributed to lack of chordwise balance and the three-per-revolution bending to the fact that the torsional natural frequency of the tail-rotor installation was about three per revolution. Had these conditions been in any way appreciated during the initial design phases, the part would have been designed accordingly.

The three failures just discussed fall into the design error classification, and it seems well to mention at this point that as the helicopter industry grows and more machines have been designed and have had their various peculiarities resolved, the state of the art will advance to the point where the percentage of failures originating in design will tend to decrease.

Manufacturing Errors – Referring again to Fig. 3, it can be seen that the next highest percentage of failures is attributable to manufacturing errors. Some specific cases of fatigue failure that are definitely attributable to errors in manufacture will now be discussed.

Fig. 7 is a photograph of a failed steel drag pin. This pin is located in the main rotor system and performs the function of reacting the aerodynamic drag and torque loads imposed upon the rotor blades. For this reason the loads are high and are of an oscillating nature. Since the actual loads were measured in the flight vibratory stress program, during the certification of the helicopter, the design was made intelligently and was quite adequate. However, due to a manufacturing error that was not detected by the inspection department, the particular part shown in the figure was accepted and installed on a production rotor system with *no* fillet radius at the change in cross-section. A very high stress concentration then existed and the part failed in 1200 hr of service.

Another example of failure due to manufacture is shown in Fig. 8. The photograph shown is of the center section of a cantilever beam, which operates

Fig. 7 – Fatigue failure of main rotor drag pin

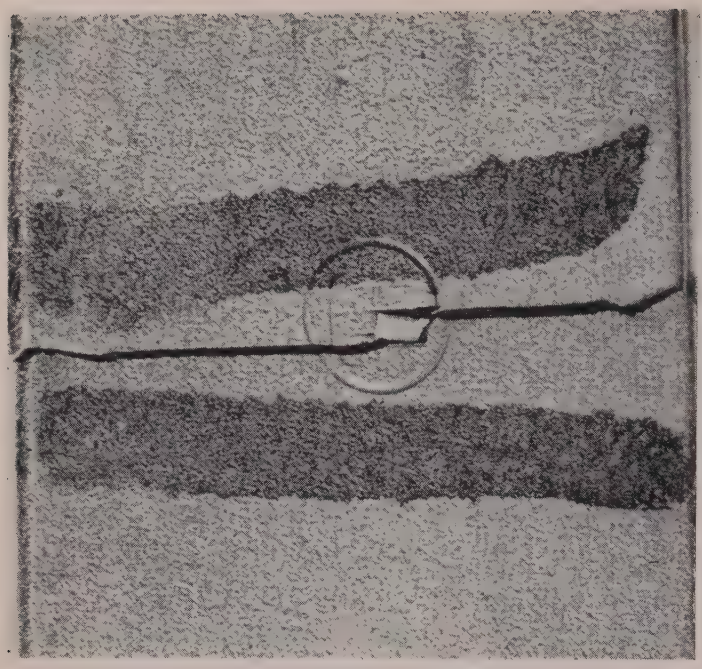


Fig. 8 – Fatigue failure of cyclic control vibration damper arm through inspection stamp

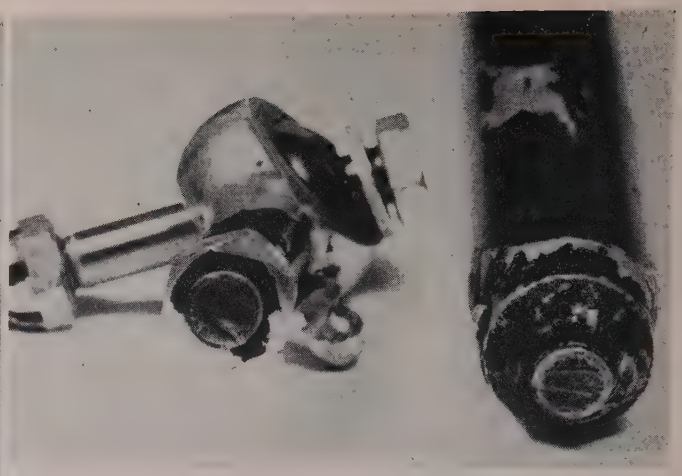


Fig. 9 – Fatigue failure of cyclic control push-pull rod assembly

as a vibration damper in the cyclic control system. The failure took place, not at the root or clamped end of the part, but at a section half-way out to the tip. As is indicated, the crack is coincident with the metal stamp impression placed on the part by the inspector. A detailed examination of the part showed that the crack actually had its origin at the steel stamp impression.

Field Errors – Reference is again made to Fig. 3 for consideration of the fatigue failures attributable to errors occurring in operation or in the field. A typical example of such a failure is shown in Fig. 9. The part shown is a failed cyclic control push-pull rod and the failure took place where the rod end bearing attaches to the plugged tube. The control loads in this part are low; however, the rod is located in such a position as to make it a very convenient hand hold to be used by maintenance personnel. Such was the case, and the failed rod was found to have a considerable amount of permanent set prior to the fatigue failure. Of course, the eccentricity caused by the (initial) bending set in the rod allowed the in-flight axial loads on the rod to produce alternating bending stresses in the rod end shank. This brought about the ultimate fatigue failure. There is no question that this failure was directly caused by the maintenance personnel. It should be pointed out, however, that the opportunity was provided by the

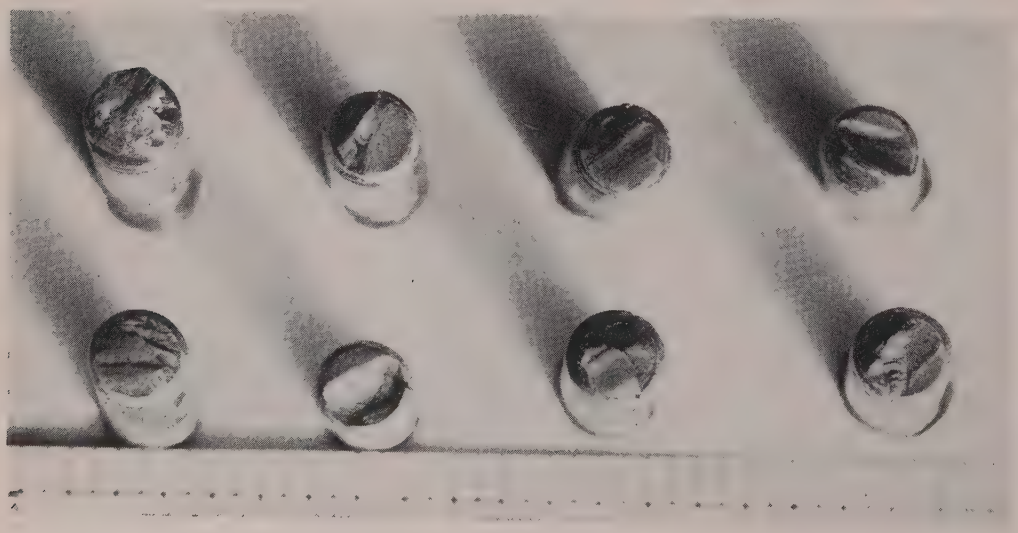


Fig. 10 – Fatigue failure of flywheel to mercury clutch attachment studs

designer and, as such, could be considered as a failure originating in the design stage.

Fig. 10 is a picture of eight studs that quite obviously failed in fatigue. These studs attach the mercury clutch to the engine flywheel and transmit the full engine torque from the flywheel to the clutch. Their failure is an interesting one, considering its origin. The following is a direct quote taken from the service representative's report on the incident. "The mechanic reports that at the time of the engine change, the nuts on the subject stud were torqued by the pilot. The pilot did not understand the workings of the torque wrench. It is the type that clicks when the set torque is reached. The pilot was of the opinion that the wrench released when the correct torque was reached. As the last nut was being tightened, the stud broke. This stud was replaced. There is no way to tell how much torque was applied to the nuts at that time." The studs were apparently all overtorqued, one actually to the point of failure. The other studs, however, went into operation under a very high initial steady stress, perhaps to the point of starting small cracks, and failed after a relatively few hours of operation. This, of course, is an excellent example of a failure caused by an operational error, since these studs normally have an indefinite life.

Experience Through Testing

Throughout the four years of operation of the model 360, a continual fatigue testing program has been in progress. The primary purpose of the program has been to improve the service life of a number of the helicopter's component parts; however, the specific nature of the tests did not prevent them from becoming a valuable source of general fatigue information. This information deals with such items as design practice, material behavior, and testing technique. For the most part, the tests merely confirmed what is currently considered good practice, while some tests produced results that were considered as contributing to the "state of the art."

Detail Design – From the standpoint of detail design, two particular bench fatigue tests are of interest. As a result of numerous operational failures, the cyclic vibration damper arms (see arm in Fig. 8) were bench tested to obtain an *S-N* curve. Different geometrical configurations were considered and tested, but, in the final analysis, it was a shotpeening operation on the original damper arm that provided the required service life. Fig. 11 presents the *S-N* curves for the original damper arm and the shotpeened arm. In the operating range the fatigue life is increased 10 to 15 times by the peening process. Fig. 12 shows the fracture of a standard damper arm and a shotpeened arm. The plain arm exhibits a failure that started at one or perhaps two points, while the shotpeened arm indicates failure starting at numerous points as evidenced by the many "beach marks." This would seem to substantiate the claim that the shotpeening process minimizes surface defects.

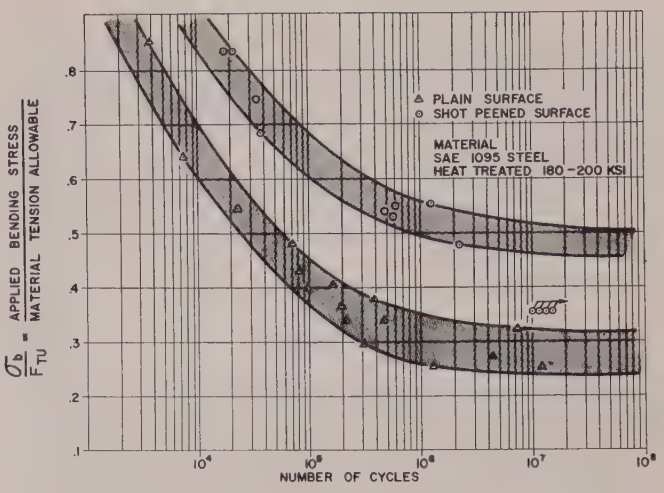


Fig. 11 – *S-N* curve for cyclic control vibration damper arms for plain and shotpeened parts

A second fatigue test, which is of interest from a detail design standpoint, concerns the control rotor cuff (cuff can be seen retaining the control rotor in Fig. 1). Fatigue tests were initiated, not as a result of service failure, but to investigate the adverse effects of a reentrant corner, which was inadvertently included in a large production order. Four specimens were bench tested at the operating stress level, and all four failed through the bolt holes as shown in the top half of Fig. 13. Examination of the part indicated that the holes were located on the forging lap line, and that failure occurred in the lap line area. Coinciding with the

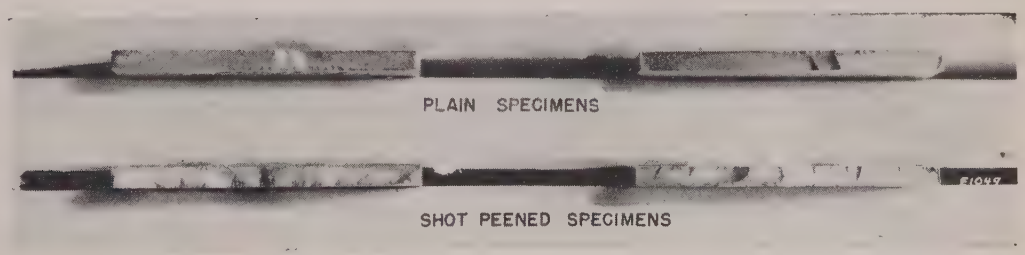


Fig. 12 – Fatigue failures of cyclic control vibration damper arms showing effects of shotpeening on failed surface

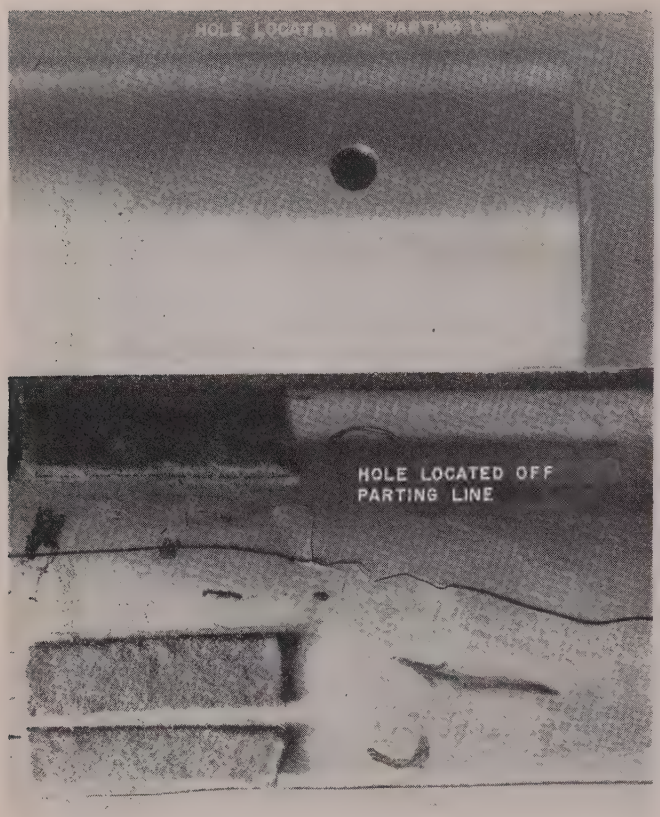


Fig. 13 – Fatigue cracks in control rotor cuffs with hole located both on and off forging parting line

conclusion of the first four tests were three field reports indicating service failures through the outboard bolt holes and in the lap line. Additional tests were then undertaken with the bolt holes rotated 45 deg from the lap line area. Two of these specimens have thus far failed, not through the lap line, but circumferentially, as shown in the bottom half of Fig. 13. Fig. 14 shows the increased life obtained by removing the stress-raiser from the lap line area.

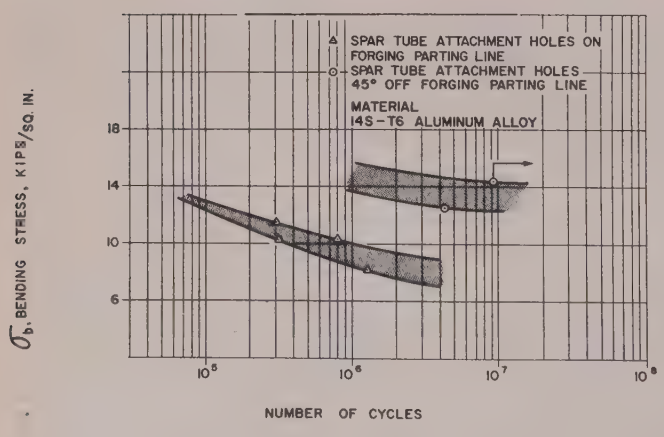


Fig. 14 – *S-N* curve for control rotor cuffs for parts with and without hole on forging parting line

Materials – Two rather interesting facts concerning material behavior have been obtained as a result of operational and fatigue tests. The first, which is a very useful observation, concerns the fatigue resistance of the high-temperature metal, Inconel X. A good deal of service and whirl test time has been "logged" on tip-mounted, ramjet engines manufactured from Inconel X. Failures have been observed, such as the one shown in the upper half of Fig. 15, where the aft tail cone has fractured in operation. Analytical and experimental studies indicated that the natural frequency of the tail-cone ring was approximately 270 cps. A forcing frequency, present during engine test operations at abnormally high fuel flows, was determined to be about 200 cps. This established the presence of periodic stresses, but also seemed to indicate that they were of a relatively low magnitude. The metallurgical report, however, furnished a definite clue to the cause of failures; the clue being the presence of lead in the grain boundaries of the parent metal. Lead carbonate was found on the interior surface of the engine as a natural product of combustion. The metallurgical examination pointed out that, at a temperature of approximately 1325 F, the lead carbonate underwent a chemical change to lead oxide and then proceeded rapidly to penetrate the grain boundaries. (See lower half of Fig. 15.) It was the opinion of the metallurgist that the lead attacked the grain boundaries and resulted in a form of corrosion fatigue; this, coupled with the low periodic stresses, resulted in final fracture. At present, ceramic coatings, shotpeening, and structural revisions are being investigated as solutions to the problem.

A second item concerning material behavior deals with the fatigue and instantaneous fracture zones of AISI 316 stainless steel. The observation is of purely academic interest and has little if any practical application. In the course of a number of fatigue tests, which were being conducted on tail-rotor tension-torsion bars (commonly referred to as "T-T" bars), considerable controversy developed concerning the location of the fatigue zones on the fracture face. Normally, the smooth "silky" surface represents the fatigue zone, while the instantaneous fracture zone has a rough and grainy surface. The condition of the surfaces did not coincide with the anticipated surface orientation due to the applied loads. The loading was a large steady tension and alternating torsion, and it was suspected that instantaneous fracture would occur on a diagonal shear plane; yet, this plane seemed to have all the characteristics of the fatigued surface. Fig. 16 shows specimens 1 and 2 which failed on the test stand. Specimen 3 was stopped with a small surface crack and fractured statically with an increased tensile load. It is apparent that in this

particular case the final rupture zone is the smooth surface and the rough grainy surface the fatigue zone.

Testing Techniques – Utilizing actual bench tests to predict service life of components requires that the particular part be carefully considered from the standpoint of the dimensional tolerances of the production item. If the part is to be subjected to simulated operational conditions, or if an *S-N* curve is to be obtained, the specimens should be chosen so as to represent the most critical combination of dimensional tolerances. Of course, in many instances the effect of tolerance range on stress level is insignificant, but in others it is extremely important. The fatigue tests that were conducted on the elliptical "T-T" bars, previously mentioned, are an example of a case where these tolerances were extremely important.

Four "T-T" bars were tested using a cyclic unit method, per Civil Aeronautics Manual No. 6, and the service life was based on the bar that withstood the minimum number of cyclic units. The testing procedure involved applying two levels of tensile load and twisting the bar periodically through a constant angular deflection. It was decided upon the conclusion of the testing of the four bars that four additional bars, in excess of the CAM-6 requirements, would be tested. Each of the latter four bars failed at a lower number of cyclic limits than any of the first four.

As a consequence of this extreme scatter of results, the eight specimens were all inspected and found to be well scattered through the tolerance range (major axis $= 0.318 \pm 0.03$ in.; minor axis $= 0.081 \pm 0.003$ in.). An investigation was then made using the approach outlined in Fig. 17, whereby a torsional stress (predominate) is calculated, based on nominal dimensions, and is then allowed to take on the tolerance extremes. It was found that a differential of about 10% in the stress level could be obtained on this particular part. This represented a possible life differential of about 100% and explained, to a great extent, the scatter of the results obtained. It is interesting to note, for this particular constant deflection test, that the

Fig. 15 – Failure in ramjet engine caused by lead penetration into material grain boundaries

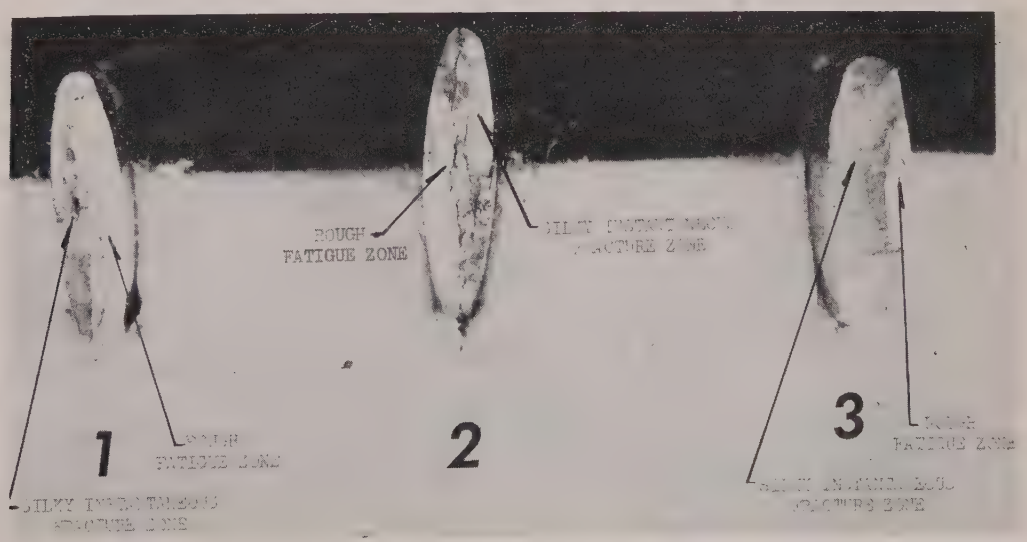
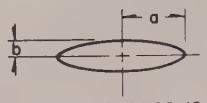


Fig. 16 – Fatigue failures of tail-rotor tension-torsion bars showing fatigue zone texture



MAXIMUM SHEAR STRESS IS $\tau = \frac{2M}{\pi a b^2}$

UNIT TWIST IS $\theta = M \frac{a^2+b^2}{\pi a^3 b^3 G}$

FOR FATIGUE TEST UNDER CONSTANT ANGULAR DEFLECTION

$$\tau = (2\theta G) \frac{a^2 b}{a^2+b^2}$$

ASSUME $(2\theta G)$ IS CONSTANT $= \lambda$

$$\tau = \lambda \frac{a^2 b}{a^2+b^2}$$

THE EXPRESSION FOR TOTAL STRESS ON THE SECTION MAY BE WRITTEN AS

$$\tau = \tau_o + \frac{\partial \tau}{\partial a} \cdot \delta a + \frac{\partial \tau}{\partial b} \cdot \delta b$$

Fig. 17 – Tolerance analysis on elliptical cross-section

rate of change of stress with major and minor axis is positive. This, of course, indicates that the most critical specimens would be those with the largest dimensions.

The problem of designing testing apparatus is a complete field in itself and the fatigue life of the apparatus acquires an importance approaching that of the component to be tested. Perhaps the experience gained in this respect can best be noted by simply making reference to a recently completed rotor-blade fatigue testing machine. (See Fig. 18.) This testing machine was designed primarily for testing teetering rotor blades and as such is capable of imposing on the specimen steady flapwise and chordwise root moments, alternating chordwise and flapwise root moments, and steady centrifugal force; all with a minimum power expenditure. It is not limited to rotor blades, but can, and will, be used on other components to supplement present experimental and service experience.

Conclusions

The subject matter presented in this paper represents a selected portion of a larger survey that we carried out. As a result of this study it was concluded that the company was faced with two major tasks in connection with the service life of its product. First, it must make an additional effort to protect the service life now enjoyed by its production machine, and second, it must initiate and carry out a more rigorous program of life extension for present and future designs. The limited information available from the service and testing of the model 360 seemed to indicate certain lines of action that could be taken to accomplish these two tasks.

Regarding the first task, a lecture and training course was set up to acquaint the inspection and manufacturing personnel with some of the pertinent facts concerning fatigue failures. This is, of course, not possible with all of the operators, but similar lectures have been introduced into military service orientation courses for maintenance personnel.

Efforts directed toward the second task have been more extensive. Fatigue design information is constantly disseminated to design groups, flight test investigations have been expanded in scope, and large expenditures have been made to improve testing technique and secure new testing apparatus.

The results of these efforts have already started to appear and seem to support the contention that the "state of the art" can be advanced considerably by the interchange and dissemination of service and testing experience.

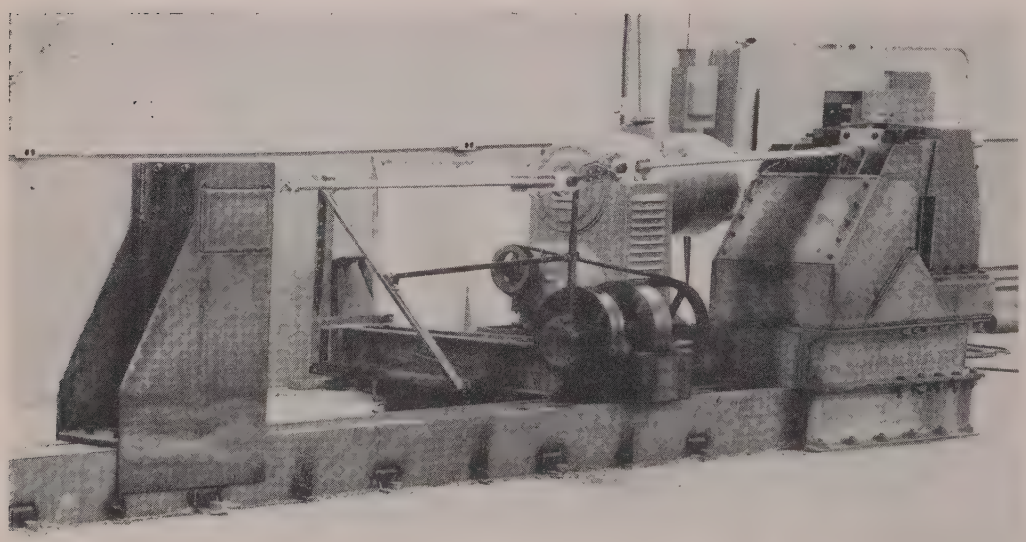


Fig. 18 – HJ-1/YH-32 rotor blade fatigue test jig

Some Effects of Motor Oils and Additives on Engine Fuel Consumption

Carl W. Georgi, Quaker State Oil Refining Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

THE friction-reducing properties of lubricants seem to have intrigued physicists and chemists, and possibly even the ancient alchemists, ever since some prehistoric experimenter happened to find that animal fat kept his chariot wheels from squeaking.

A cursory review of the literature reveals literally hundreds of papers on friction measurements in an almost bewildering variety of laboratory apparatus. Unfortunately, published data on the correlation of laboratory friction measurements with lubrication and friction reduction in commercial machines and engines is very meager.

While laboratory technicians have long been active in studying friction, the ones who have really gone all out are the boys who write advertising copy for some of the commercial motor oils and motor-oil additives. In recent years, we've seen such advertising slogans as:

"Makes oil 300% more efficient."
"Reduces friction by 75%."
"Saves 1 gallon of gasoline in every ten."
"Increases power by 10%."
"8% more power and 3 more miles per gallon."

Some of the more conservative sloganeers content themselves with less specific phrases such as: "decreases friction," "increases engine power," or "increases gas mileage."

Since friction, power, and fuel economy have become such popular bywords in connection with engine lubrication, and since no published engine test data seem to be available, we thought it might be of some interest to explore the subject in an engine. This idea was further advanced by certain rumors we have heard that engines are frequently quite unimpressed by tests in laboratory apparatus or by the claims of advertising copy writers.

In considering friction reduction through lubrication from an engine's viewpoint, it would appear that if any given oil does have superior friction-reducing properties, such would readily be reflected by an increase in engine power output at any fixed conditions of engine speed and rate of fuel consumption. Conversely, at any set conditions of engine speed and power output, an improvement in fuel economy should become evident. Since fuel economy or "miles per gallon" is the item of great-

MANY rather intriguing claims have been made about reduction of engine friction and increase of power and fuel economy through the use of certain types of motor oils and motor-oil additives.

However, this paper shows that engine fuel-consumption measurements to determine the effects of different oils and oil treatments indicate that the only significant property of the engine lubricant relating to engine power, friction, and fuel economy is viscosity. The data also indicate that the "engine viscosities" of some motor oils may be somewhat different from their "laboratory viscosities" as measured in conventional Saybolt or Kinematic viscosimeters.

The Author

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Table 1 – Test Oils

Oil No.		Viscosity Index	Saybolt Viscosity, SUS 210 F	Saybolt Viscosity, SUS 300 F	Additive Treated "for Service"	Viscosity Index Improver
1	High V. I. SAE 5W	103	39.0	32.6	MS-DG	No
2	High V. I. SAE 10W	110	47.8	36.1	MS-DG	No
3	High V. I. SAE 20-20W (Reference Oil)	110	58.3	39.4	MS-DG	No
4	High V. I. SAE 30	107	68.2	42.0	MS-DG	No
5	High V. I. SAE 50	105	94.1	47.8	MS-DG	No
6	50 V. I. SAE 10W	48	39.8	32.6	ML	No
7	50 V. I. SAE 20W	49	50.0	36.0	ML	No
8	50 V. I. SAE 30	53	61.0	38.8	ML	No
9	50 V. I. SAE 50	57	87.8	44.3	ML	No
10	0 V. I. SAE 10W	10	40.3	32.6	ML	No
11	0 V. I. SAE 30	−11	66.0	39.0	ML	No
12	High V. I. SAE 20W plus Colloidal Graphite Additive	112	55.1	38.4	MS-DG	No
13	High V. I. SAE 20W plus Moly Disulfide Addi tive	110	57.3	39.0	MS-DG	No
14	High V. I. SAE 20W plus Lead Soap Additive	115	51.0	37.3	MS-DG	No
15	SAE 5W-20 "A"	151	44.8	35.7	MS-DG	Yes
16	SAE 5W-20 "B"	154	50.2	38.1	MS-DG	Yes
17	Experimental SAE 5W-20	161	58.5	42.0	MS-DG	Yes
18	Experimental SAE 10W-30	153	67.0	44.9	MS-DG	Yes

est interest to most motorists, this test program was set up on that basis.

A popular make of passenger-car engine was mounted on a dynamometer for conventional fuel-consumption measurements under a variety of speeds and loads and with a variety of motor oils. The usual precautions were taken both with the engine and the operating procedure to insure optimum uniformity of test conditions. Tests on each oil were run at engine speeds equivalent to 15, 30, 45, and 60 mph on the road, and at loads of 25%, 50%, 75%, and 100%. This range of engine speeds and loads was believed to cover reasonably well conditions ranging from part-throttle level-road operation to wide-open-throttle "lugging" or hill climbing. Each oil was accordingly tested under 16 different load-speed combinations. Check fuel consumption readings were taken at each setting, so that a minimum of 32 consumption readings was made in each test run.

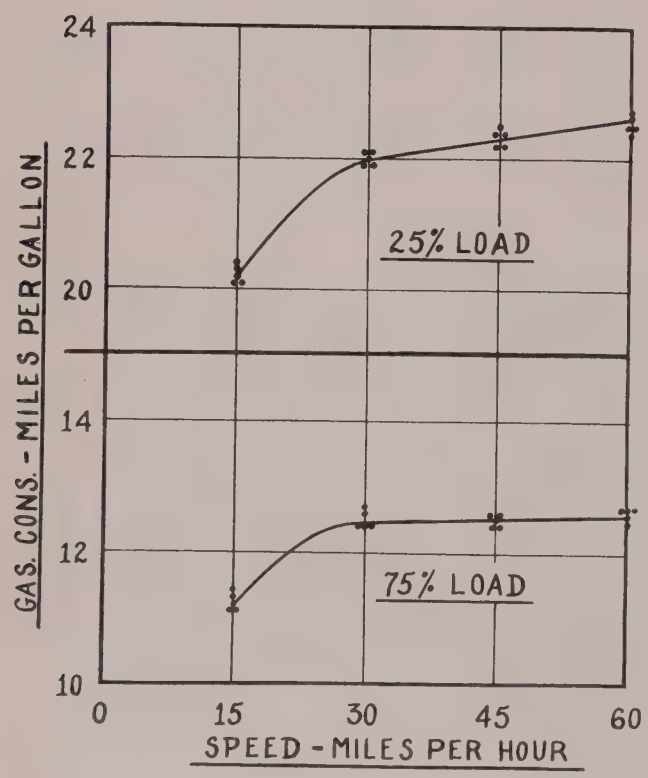

Fig. 1 – Typical effects of engine load and speed on fuel consumption

Table 1 lists the more pertinent properties of the motor oils tested. All of these were commercial products, with the exceptions of the first and last two oils listed. The test oils comprised five of naturally high viscosity index in the SAE 5W through SAE 50 grades; four oils of 50 V.I. in SAE 10W through SAE 50 grades; and two oils of 0 V.I. in SAE 10W and SAE 30 grades.

Three of the test oils consisted of the SAE 20 high V.I. oil treated with three popular crankcase-oil additives. One of these additives contained colloidal graphite as the "active" ingredient, the second molybdenum disulfide, and the third lead soap. The treatment in each case consisted of one can of the additive per crankcase fill with the SAE 20 reference oil, per the directions contained on the cans of the respective products.

The four oils of 150 plus viscosity index were all heavily doped with V.I. improver additives.

The SAE 20 high V.I. oil was used as the reference oil. Every fourth or fifth test run in the engine was made with this reference oil, to make certain that the engine and its operation remained constant, and that no undetected changes developed during the course of the program to compromise the overall test results. Fig. 1 illustrates typical results from five such check tests on the reference oil. Check tests at all speed-load conditions agreed within plus or minus 0.2 mpg, which indicates the accuracy and reproducibility of the test results.

It will be noted from Fig. 1 that fuel consumption in "miles per gallon" varied markedly with engine load but varied only slightly with engine speed over the 15 to 60 mph range. At each test load, fuel consumption was somewhat higher at the low speed of 15 mph than at the higher speeds. This is to be expected, since most carburetors deliver richer mixtures at low engine speeds.

The tests on 18 different oils, each under 16 different engine speed-load combinations, obviously resulted in a large mass of fuel-consumption figures. In order to summarize these data in as simple a form as possible, the average fuel consumption at the four test speeds was figured for each oil at each of the four test loads. As shown in Fig. 1, this simplifying step could be taken, since "miles per gallon" varied relatively little with engine speed but varied materially with load.

Fig. 2 shows these average fuel-consumption values plotted against oil viscosity at 210 F. The

points plotted with dots represent the five high V.I. oils; those plotted with circled dots, the SAE 20 reference oil containing the three different proprietary additives; the squared dots, the low V.I. oils; and the "X's," the oils containing V.I. doping.

The first item of interest is that oil viscosity has a more significant effect on fuel economy than does oil composition. This is in accord with theoretical considerations, since the internal friction of oil films naturally increases with increase of oil viscosity.

The second item of interest is that the three proprietary crankcase additives showed no differences in fuel consumption from that of the motor oil to which they were added, within the limits of test reproducibility. This seems to be rather clear evidence that the engine did not recognize any friction-reducing properties in graphite, "moly" disulphide, or lead soap.

The third item of interest is that all of the low V.I. oils consistently caused more fuel consumption than the high V.I. oils on an equivalent 210 F viscosity basis. The differences are more pronounced at the lower engine loads.

The fourth item of interest is that all of the oils containing V.I. improvers consistently developed less fuel consumption than the other oils on an equivalent 210 F viscosity basis. The differences become more pronounced at lower engine loadings, as well as with higher oil viscosities.

A number of investigators have suggested that oil viscosities at 300 F may correlate more nearly with various factors involved in engine lubrication than do viscosities at 210 F. Fig. 3 accordingly shows the average fuel-consumption values plotted against oil viscosity at 300 F. The plots of fuel consumption against oil viscosities at 210 and 300 F are in complete agreement in all respects. The only significant item is that the low V.I. oils, as well as the oils containing V.I. improving additives, show even larger deviations from the median in the 300 plot than they do in the 210 F plot.

Since the spread of results with the different types of oils seems to increase with increase of temperature selected for comparing oil viscosities, another plot was made of fuel consumption versus viscosities at 150 F, as shown in Fig. 4. The data again show the same trends, although the deviations are very small.

However, during the fuel-consumption measurements, engine water-jacket temperatures were maintained at 150 ± 5 F, and crankcase-oil sump temperatures ranged from 160 to 270 F, depending on the engine speed and load and on the viscosity of the oil. It is believed, therefore, that oil-film temperatures within the engine averaged much higher than 150 F, and that the plots of fuel consumption against oil viscosities at 210 or 300 F are more indicative of the effects of different types of oil on fuel consumption.

These fuel-consumption data indicate the extent of fuel savings to be attained with lighter grades of oils. Table 2 summarizes the data in the form of fuel-consumption ratios as related to oil viscosity grade. Use of a straight SAE 5W motor oil of minimum 210 F viscosity provided an increase of 3% to 6% in "miles per gallon" compared to a

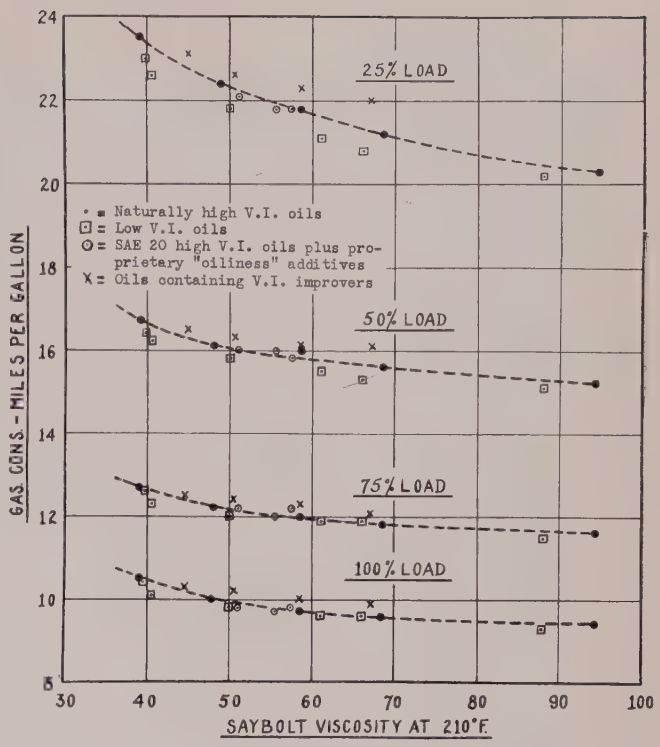

Fig. 2 – Average fuel consumption versus motor-oil viscosity at 210F

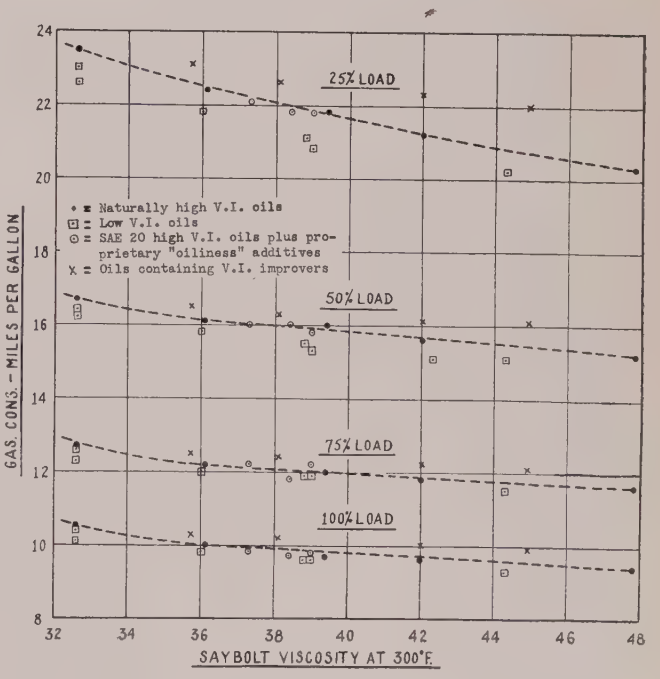

Fig. 3 – Average fuel consumption versus motor-oil viscosity at 300 F

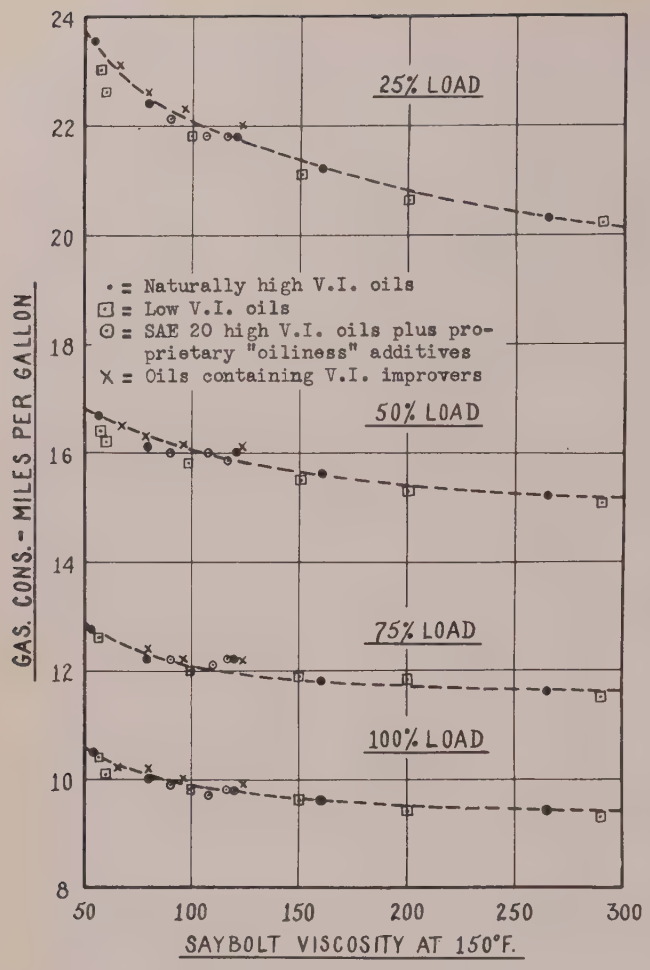

Fig. 4 – Average fuel consumption versus motor-oil viscosity at 150 F

typical SAE 20 or 20W grade. Use of a typical SAE 50 grade caused a decrease of 5% to 7% in "miles per gallon" compared to a typical SAE 20 or 20W grade. On the average, a change by one SAE viscosity grade thus produced a change of around 2 or 3% in "miles per gallon."

While these indicated savings in fuel consumption with lighter oils appear to be quite significant, they apply only to "ideal" laboratory conditions with very close control of engine operation and precision measurement of fuel consumption. On the road, it is doubtful if fuel-consumption differences of this order could be detected by the average motorist. Tank mileage on the road (the number of gallons of gasoline purchased divided by the speedometer mileage) is vitally affected by the driving conditions. As an example, a typical road-test car over a period of time showed average tank mileages of 12 mpg in city driving in winter, 15 mpg in warm-weather city driving, and 19 mpg in cross-country operation.

The large variations in "miles per gallon" caused by type of operation would certainly tend to obscure the relatively small variations resulting from motor oils of different types or viscosities. The isolated effects of motor oils on engine fuel economy thus appear to be essentially of technical interest, and they do not appear to be of appreciable practical significance as far as motorists are concerned.

Table 2 – Typical Fuel-Consumption Ratios as Related to Oil Viscosity Grade

SAE Viscosity Grade	Gasoline-Consumption Ratio
SAE 5W	1.03 to 1.06
SAE 10W	1.01 to 1.03
SAE 20-20W	1.00
SAE 30	0.96 to 0.98
SAE 50	0.93 to 0.95

These fuel-consumption studies also introduce two questions:

1. Why did the low V.I. motor oils indicate higher than average fuel consumption?
2. Why did the oils containing V.I. improving polymers indicate lower than average fuel consumption?

Again, while these differences were of sufficient magnitude to be of technical interest, it is doubtful if they could be detected on the road by motorists.

The low V.I. oils performed, in terms of fuel consumption, as if their "engine viscosities" were quite a bit higher than their viscosities as measured in conventional laboratory apparatus. A possible explanation which might thus be advanced relates to the effects of pressure on oil viscosity. Just as oils change in viscosity with temperature, so they increase in viscosity with increase of pressure. Low viscosity index oils are much more sensitive to viscosity change with both temperature and pressure than are high V.I. oils. Even at the relatively moderate pressures developed on oil films in engines, low V.I. oils could conceivably undergo a sufficiently greater pressure-viscosity increase to account for the higher than average fuel consumption.

This is illustrated by Table 3. The three oils of 10, 108, and 152 viscosity index and of similar viscosity at 210 F at atmospheric pressure, developed significant differences in viscosity at 2000 lb pressure and 210 F. It seems possible that the much greater pressure-viscosity increase of the low V.I. oil, at pressures within the range which might be expected to be imposed upon oil films in engines, could account for the fuel-consumption data.

The very high V.I. oils containing polymer additives performed, in terms of fuel consumption, as if their "engine viscosities" were appreciably lower

Table 3 – Effects of Pressure on Oil Viscosity

	Viscosity Index	Viscosity at 210 F, centipoises: Atmospheric Pressure	Viscosity at 210 F, centipoises: 2000 Psi Pressure	Per Cent Viscosity Increase
Naphthenic Oil	10	10.1	18.6	84%
Paraffinic Oil	108	10.2	13.4	31%
Paraffinic Oil plus V. I. improver	152	10.1	12.9	28%

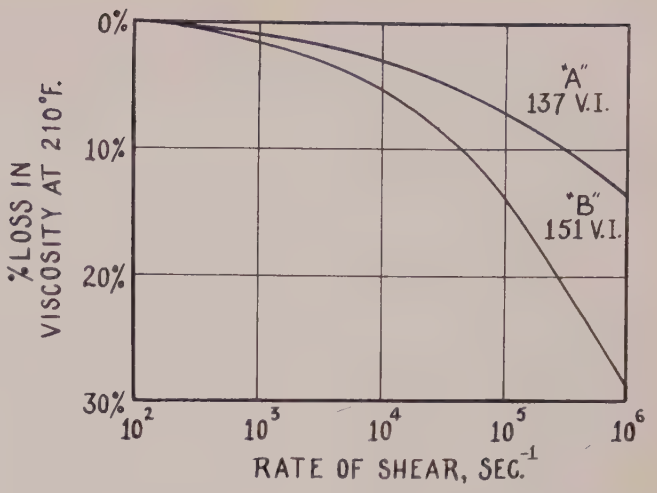

Fig. 5 – Typical effects of shear rate on oils containing V. I. improvers. Data are from tests in tapered-plug viscosimeter-oil A: 50 SSU at 210 F; oil B: 54.9 SSU at 210 F

than their conventional laboratory viscosities. It seems reasonable to assume that this difference was largely due to the polymer additives. Since the pressure-viscosity characteristics of oils containing V.I. improvers do not seem to be greatly different from that of the base oils alone, the pressure factor does not appear to account for the lower fuel-consumption values. Another explanation which might be advanced relates to the fact that V.I. doped oils are not Newtonian fluids. That is, their viscosities are affected by rate of shear, as well as by temperature and pressure.

Fig. 5 illustrates some typical effects of shear rate on oils containing V.I. improving polymers. Oil "A" had a viscosity of 50.0 SUS at 210 F and contained sufficient polymer additive to have a viscosity index of 137. Oil "B" had a viscosity of 54.9 SUS at 210 F and a larger amount of polymer additive to attain a viscosity index of 151. It will be noted that viscosity loss becomes very appreciable at shear rates above about 10,000 reciprocal sec, and that the viscosity loss is greater with the oil containing the larger amount of V.I. improving polymer. Since shear rates imposed on oil films in engines can readily exceed 10,000 reciprocal sec, it seems reasonable to conclude that the non-Newtonian characteristics of V.I. doped oils can account for their lower fuel-consumption tendencies.

Summary

On the basis of the engine fuel-consumption data presented, the following conclusions may be drawn:

1. Popular types of "oiliness" or "friction-reducing" additives display no measurable improvement in engine friction, power, and fuel economy over that of the motor oil to which they are added.

2. The only significant property of motor oils relating to engine power, friction, and fuel economy is viscosity.

3. "Engine viscosities" of some motor oils may be quite different from their "laboratory viscosities" as measured in conventional Saybolt or Kinematic viscosimeters, due to the effects of pressure and of shear rate on oils of non-Newtonian character.

4. Low V.I. oils are indicated to have "engine viscosities" appreciably higher than their "laboratory viscosities," possibly due to the fact that low V.I. oils are much more sensitive to viscosity increase with increase of pressure.

5. Oils containing V.I. improving polymers are indicated to have "engine viscosities" appreciably lower than their "laboratory viscosities," probably due to the fact such oils are non-Newtonian and develop considerable loss of viscosity at high rates of shear.

6. It is particularly interesting that these fuel-consumption tests indicate the "engine viscosities" of oils containing V.I. improvers are lower than their "laboratory viscosities," since engine tests on the oil-consumption tendencies of such lubricants indicate the same condition.[1] Two different approaches, one by engine tests on fuel consumption and the other by engine tests on oil consumption, thus indicate the same conclusion; namely, that oils containing V.I. improving polymers have "engine viscosities" appreciably lower than their "laboratory viscosities."

[1] See SAE *Quarterly* Transactions, Vol. 3, July, 1949, pp. 431-443: "Motor Oil Consumption Characteristics," by C. W. Georgi.

DISCUSSION ON FOLLOWING PAGE →

Some V.I. Improved Oils Will Up On-the-Road Mileage

– H. F. Galindo and J. A. Miller
California Research Corp.

WE are indebted to the author for his praiseworthy endeavor to limit advertising to claims which can be technically justified. Obviously the "reduction in friction" claims have been overdone and in many cases are completely unfounded. We feel, however, that certain legitimate technical advantages have been included by the author in the unfounded category. As Mr. Georgi states, the engine is an excellent critic of the performance of a fuel or lubricant. However, he might have gone one step farther and stated that the engine, in the hands of the average driver, under average driving conditions, is the final board of review. There are several important factors missing in the otherwise excellent and precise laboratory engine results presented by Mr. Georgi. His engine and dynamometer stand were not equipped with a traffic light or a parking lot. Even in the Far West we find that the number of "wide-open spaces" to drive a car at high speeds and loads are diminishing rapidly, and traffic driving has become a major factor affecting gasoline and lubricating-oil mileage.

We would like to present some additional data with regard to the gasoline mileage advantages obtained with V.I. improved oils. Fig. A illustrates some gasoline mileage data obtained in a passenger car, in average mechanical condition, under typical city driving conditions, with the normal amount of stopping, starting, and traffic lights. These tests were made with a specially compounded, V.I. improved oil, meeting the requirements of a 10W oil at 0 F and an SAE 30 grade oil at 210F. The reference oil was a normal SAE 30 grade, heavy-duty-type oil. The individual checks on gasoline mileage with each oil were consistent, and the average gain in gasoline mileage on this traffic run was approximately 14% in favor of the 10W-30 oil. Fig. B shows a similar test with a secretary driving the car. The driver had no instructions with regard to the handling of the throttle, and she was advised to drive in her normal manner. This test was conducted over a longer trip with less stops involved and with higher speeds and temperatures prevailing throughout most of the run. Under these conditions a 9% gain in the fuel mileage was obtained with the 10W-30 V.I. improved oil.

In order to determine if these effects could be found in passenger-car operation with different drivers and with gasoline mileage checked only with the service-station pump and speedometer readings, a driving pool of 11 cars was used for a gasoline mileage test. The results are listed in Table A. Various makes of cars were involved, and many drivers used these cars indiscriminately. Oils were alternated on a monthly basis so each car used both oils. Under city-traffic driving conditions the overall fleet

Table A – Effect of Crankcase Oil on Gasoline Mileage of Various Passenger Cars, City-Traffic Driving

Car	SAE 10W-30 Oil, mpg	SAE 30 Oil, mpg	Relative Gasoline Mileage
A	7.8 11.6	5.9	164
B	10.4 10.8	10.6	100
C	10.5 18.0	8.3	172
D	9.8 9.9	10.7	93
E	12.7 16.5	14.9	98
F	12.7 13.6	13.1	101
G	12.0 14.0	10.9	119
H	12.6	11.3	112
I	15.7 18.2	14.5	117
J	10.1	9.1	111
K	13.5 16.6	14.8	102
Average	12.9	11.3	114

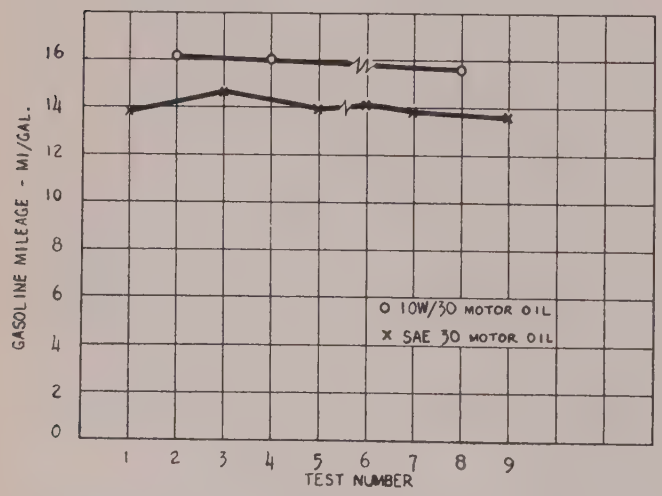

Fig. A – Effect of crankcase oil on gasoline mileage for passenger car in city-traffic driving

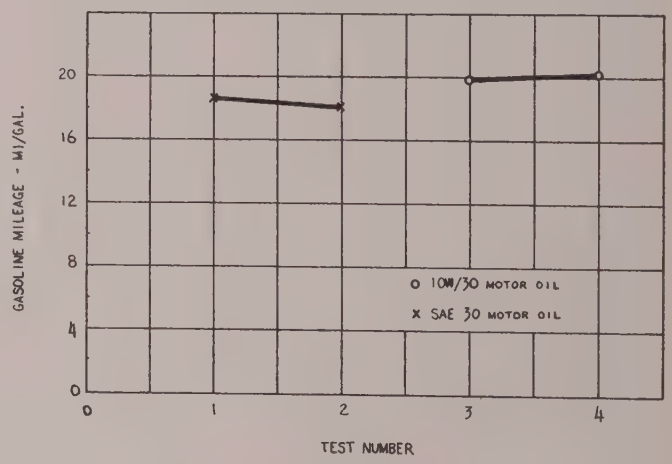

Fig. B – Effect of crankcase oil on gasoline mileage for passenger car with secretary driving in city traffic

average was 11.3 mpg with the SAE 30 grade oil and 12.9 mpg using the 10W-30 oil with an overall 14% improvement in gasoline mileage in favor of the 10W-30 oil. Of the 11 cars, only two of the cars showed a slight decrease in gasoline mileage using the 10W-30 grade oil.

In summary, we agree with Mr. Georgi's conclusion that "engine viscosities" are the primary factor influencing instantaneous engine power, friction, and fuel economy. However, the magnitude of the fuel savings effected with certain types of V.I. improved oils has not been evaluated by Mr. Georgi in terms of the type of driving that many motorists are accustomed to. As a matter of fact, we do not agree with Mr. Georgi that the 6% savings he obtained in his dynamometer tests would not be of practical significance to the average motorist. We have found a surprising response from the motorist with respect to checking gasoline consumption. Advertising claims are usually believed by the average motorist if they are made by reputable oil companies. However, many motorists run their own personal check tests. Again we wish to praise Mr. Georgi's efforts to assist the motoring public by emphasizing the need for reliable advertising. On the other hand, we do not feel it fair to depreciate reasonable advantages to the motorist when they can be substantiated on a technical basis.

Hopes for Future Data On Low Internal-Friction Engines

– L. J. Grunder
Richfield Oil Co.

THE conclusion that viscosity is the greatest single important factor in engine friction measurements is borne out by Mr. Georgi's tests and those of many other experimenters. The extensiveness of his data on one engine type, however, should encourage others in the future to further this information by running engines of even lower internal friction – where the viscosity factor may be of even greater importance percentagewise.

Although the laboratory work shows a definite trend toward fuel economy over the SAE 20 reference oil, road tests generally obscure that small difference, according to work we have done.

Author's Closure To Discussion

In making a reply to the discussors, I might mention first that the time allotted for the presentation of the paper necessitated that it be confined to a brief review of dynamometer tests in only one make of engine. Time did not permit presentation of considerable additional data from dynamometer test runs on two other makes of engines, as well as from road tests.

Our laboratory engine test work to study the effects of motor oils and additives on engine fuel consumption was conducted on three different engine makes, two of which were of the newer low-friction type having bore diameters greater than their piston strokes. The third engine was of older basic design, with stroke greater than bore diameter. The test data from all three engines as to the effects of oils and additives on fuel consumption were so similar that the data presented in the paper can be considered as typical for all three engines.

Messrs. Galindo and Miller make two pertinent points in their discussion; with one of which I fully agree, with the other of which I don't agree, based on our test experience. They state that engines on the road are the final "proof of the pudding," and this is obviously correct. In view of this fact, we ran extensive road tests on various types and viscosities of motor oils (and additives) to supplement our dynamometer test work. However, their road tests on 11 cars indicating anywhere from an average of 14% up to 72% greater gas mileage with a 10W-30 type of motor oil compared to a straight SAE 30 grade motor oil is completely out of line with our work. I can't help thinking that if such results were consistently true, the car manufacturers might want to consider retiring a large part of their engineering staffs and let the oil industry take over the entire job of developing "more miles per gallon" for the motoring public!

Although it is possible to show limited improvements in fuel economy with lighter viscosity oils in carefully controlled dynamometer tests, these differences are largely obscured by variations in operating conditions in actual service on the road. We note that Mr. Grunder has drawn this same conclusion. We can cite typical results from just one road test car to illustrate this important point. In steady, cross-country operation this car averaged 20.2 mpg with an SAE 30 motor oil of 105 viscosity index. Under as nearly identical operating conditions as possible to attain on the road, this car averaged 20.8 mpg with an SAE 5W motor oil of 145 viscosity index. Additional tests on oils of intermediate SAE viscosity grading produced fuel consumption values between these two extremes. Under steady cross-country operation, the maximum difference between a considerable variety of oils thus amounted to only 3% in "miles per gallon of gas."

A second series of road tests was run on a test circuit in heavy city traffic. Under these constant stop-go operating conditions, fuel consumption averaged 15.4 mpg with the SAE 30 oil of 105 V.I. and 15.6 mpg with the SAE 5W oil of 145 V.I., with numerous other oils falling in between these extremes. This gives a maximum difference of only 1.3% in fuel consumption.

This same car was then operated in typical city traffic in winter on a schedule involving start from cold and running for 8 miles, followed by parking for a minimum of 4 hr, followed by another cold start and 8-mile city drive, another cooling period, and so on. This test routine was intended to simulate cars used only for driving to work or similar intermittent, short-run city operation. Under these conditions, gasoline consumption ranged from 9 to 11 mpg with a variety of oils ranging from the SAE 5W up through the SAE 30 viscosity grades. The difference between 9 and 11 mpg (quite a percentage difference) related not to the viscosity or viscosity index of the motor oils used, but to the atmospheric temperature. The colder the weather, the poorer the gas economy under these short-run, cold operating conditions. Oil viscosity had no relation to fuel consumption, and in numerous instances the poorest gas mileage was encountered with the lightest oils. Atmospheric temperature and engine warmup time were the controlling factors.

These road test data on fuel consumption are typical of many we have run, and can be summed up that operating conditions can make a difference of over 100% in "miles per gallon" fuel consumption, whereas differences in oils as to viscosity and additive treatment rarely show more than a very few per cent difference in "miles per gallon" when compared under similar driving conditions. It's readily understandable, therefore, that uncontrolled road tests or the experiences of motorists may show quite wide variations in "miles per gallon," but these differences are indicated to relate far more to the driving conditions than to the oil used, even though the oil is sometimes rather optimistically given the credit.

What Happens to a Passenger

E. C. PICKARD[1] has given an excellent presentation of the characteristics of the average passenger who rides and drives our motor vehicles. He has shown how the cushions are loaded by the static forces of gravity as they support the average passenger. F. C. Matthaei, Jr.,[2] has shown what may be done to seat both large and small passengers in a comfortable and safe driving position. It remains for me to present a preliminary study of the action of the passenger on the cushioned seat when the car is bouncing over rough roads. Here the cushions are subjected to dynamic loads, sometimes large enough to cause the passenger to compress completely the cushions and strike bottom. Probably the most disagreeable passenger motion is caused by a cyclic movement of the car when it passes over a series of bumps, each successive bump adding to the violence of the action. The four simple movements of the car are:

1. The straight up-and-down hop.
2. The angular waggle about a vertical axis.
3. The angular roll about a horizontal, longitudinal axis.
4. The angular pitch about a horizontal, transverse axis.

While all of these movements may occur singly or in combination, the most disagreeable is the pitch which is produced by successive impacts at regular intervals on first the front wheels and then the rear wheels. The car movement is like that of a children's teeter-totter. It may be shown that the impacts, alternately on front and rear, will occur at regular intervals if the bumps are spaced at two thirds of the car wheelbase. Then, if the car is driven at the proper, comparatively slow speed, the

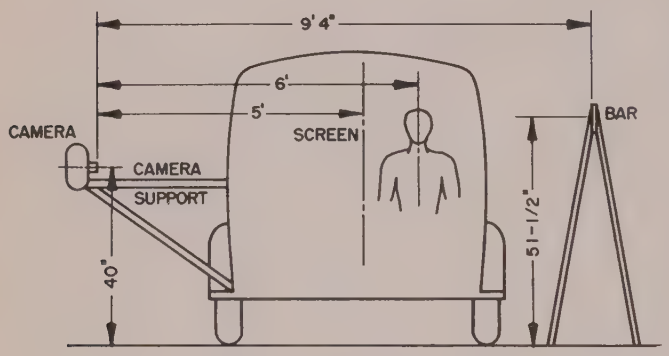

Fig. 1 – Diagram of photographic apparatus

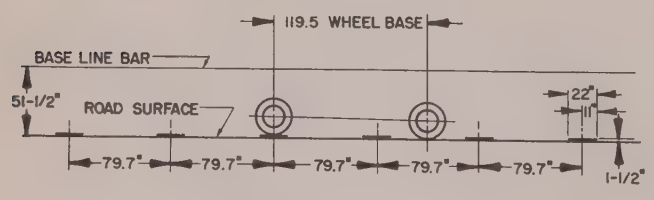

Fig. 2 – Artificial bumps on concrete road

[1] "Use of 'Oscar' in Seat Design," by E. C. Pickard. Presented at SAE Annual Meeting, Detroit, Jan. 11, 1954.

[2] See pp. 331-334 of this issue: "Universal Positioning Seat Track," by F. C. Matthaei, Jr.

When Traveling in a Car

Walter E. Lay, University of Michigan

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1954.

impacts will occur at the natural frequency of the car pitch, and each successive impact will add to the violence of the movement. This arrangement will produce large enough displacements to make possible a qualitative study by ordinary photographic methods, especially when the shock absorbers are removed.

As shown by Fig. 1, an ordinary movie camera was mounted well outside the right side of the car. With the right door removed the camera could "see," through a coordinate wire screen, the driver and a base line. The base line was mounted on horses placed along the pavement so that it was at a constant height above the pavement. The camera would then record the movement of the passenger relative to the car and also the movement of the car relative to the base line and to the pavement.

Fig. 2 is a diagram of the bumps. They consisted of sections of 1½ x 12-in. planks connected by metal straps so that they would stay in position while the car was passing over them. The car had a wheelbase of 119.7 in., and its pitch had a natural frequency of 105 cpm. The space interval between the bumps is then two thirds of 119.7, or 79.8 in. The time per cycle is 60/105 = 0.58 sec, and the car speed should be approximately 8 mph.

The test procedure included taking some pictures with the camera mounted on a stationary tripod. This included pictures showing the test area and equipment, some with the camera following the car as it traversed the bumps, and finally a timing shot of a stop watch to check the camera speed. The camera was then mounted on the car, as shown in Fig. 1, and a series of pictures made with the camera viewing the driver through the screen. (See Fig. 3.)

The driver was dressed in swimming trunks and shoes. While he was in a standing position, the principal hinge points of his body were located and marked with black crosses and the crosses connected by straight lines. The points located were:

(a) The top of the spine – base of skull.
(b) The shoulder hinge point.
(c) A point under the arm.
(d) The hip hinge point.
(e) The knee hinge point.
(f) The ankle hinge point.

When the pictures were developed, the position of each point was read from the coordinate screen

WHY one gets that aching neck and those tired eyes after driving for long stretches is explained in this paper.

A brief qualitative study of the motion of the driver and passenger in the front seat of an automobile is given here. The author provides a reasonable explanation for some of the commonly observed kinds of driver fatigue and points the way for some improvement in the dynamic comfort of automobile seats.

The Author

WALTER E. LAY (M '19) is professor of mechanical engineering in charge of the automotive laboratory at the University of Michigan. He continued on the faculty there after his graduation, but during the years has collaborated on experimental test and research work for Packard, Chevrolet, General Motors, NACA, and other similar groups. During World War II, he conducted special training courses in testing aircraft and tank engines.

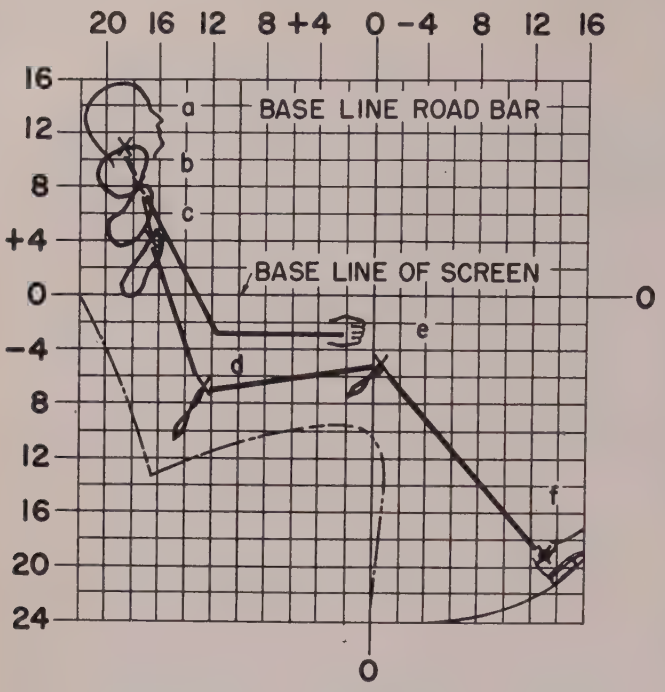

Fig. 3 – Passenger motion against coordinate screen

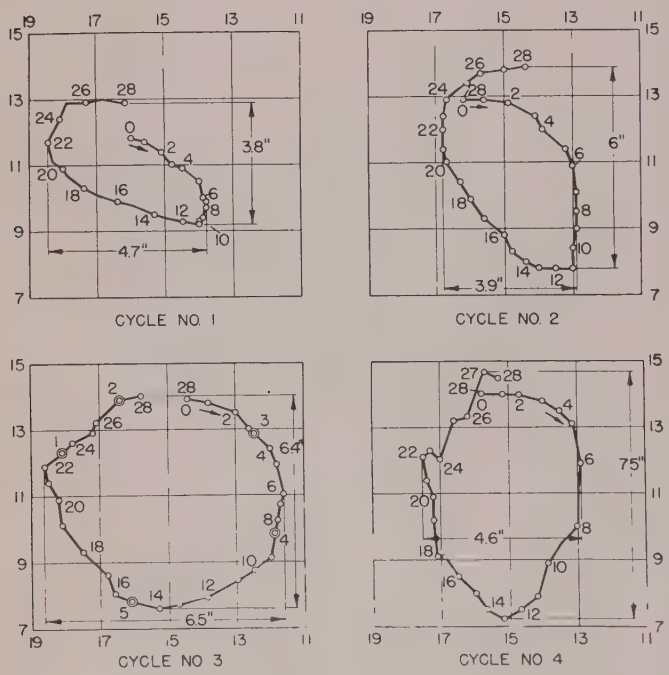

Fig. 4 – Skull movements against coordinates

for each successive frame. Knowing the distances from the camera to the point on the driver's body and to the screen, the actual position of each point was plotted. In a similar manner the absolute motion relative to the road could be determined. The type of motion of each point is:

(a) Approximately a circle.

(b) and (c) Modified ellipses.

(d) An inclined straight line.

(e) A circular arc about point (f).

On Fig. 4 is shown a plot of the actual motion of point (a), the top of the spine, through four complete cycles. It will be noted that each cycle has 28 points corresponding to 28 frames or 28/48 = 0.58 sec. As the fourth cycle was completed the car had traversed the bumps, but the pitching continued for some time on the level pavement. The motion of this fourth cycle is most typical and was verified by continuous pitching of a car on chassis drums. Five positions of point (a) on this plot were chosen, and corresponding positions of the other hinge points (b) to (f) were shown on Fig. 5 to illustrate the movement of the head, trunk, and thighs.

By connecting all of the hinge points, (a), (b), and so forth, by straight lines, the approximate position of the spine and thigh bones is shown in Fig. 5. It will be noted that in one position the spine is nearly straight, and in another it is flexed into a curve. As the shoulder point begins to move backward the upper end of the spine is still moving forward. The heavy weight of the head is rather poorly supported on the comparatively slender neck and acts much like the snapper on the end of a whip. This action may explain the ache in the neck which often results from a long rough drive. The midpoint of the spine moves backward carrying the flesh behind it deep into the seat back. The lower end of the spine moves backward into the seat back and downward into the seat cushion. As this occurs the angle between the spine and the thigh is decreased. The thigh moves downward into the cushions and also has angular motion about a point well forward of the knee. When the movement of the supporting surfaces is known, then the springs in the seat and back cushions may be proportioned so that the most comfortable pressure distribution may be preserved throughout the movement.

Another interesting observation may be noted. The whip lash action of the head and neck causes the head to rotate on the hinge point on the top of the spine. Since the eyes must always be aimed at the horizon, the eyes must be continually rocked upward and downward. This action probably causes fatigue of the eye muscles.

On some of the tests the driver guided the car up to the bumps, then removed his hands from the steering wheel while the car was traversing the bumps. His body was then free to move like that of a passenger in the front seat. The result was rather

startling in that the movement of the head and upper part of his body was nearly doubled. This explains why the front-seat passenger is more irritated by a rough ride than the driver. It indicates that the development of some kind of a hand rest for the passenger deserves serious consideration.

While this brief qualitative study is not very accurate, it may serve to rationalize some of our every-day observations. It may also indicate the direction for further investigation and help seat-cushion designers to build even a little more comfort into their product.

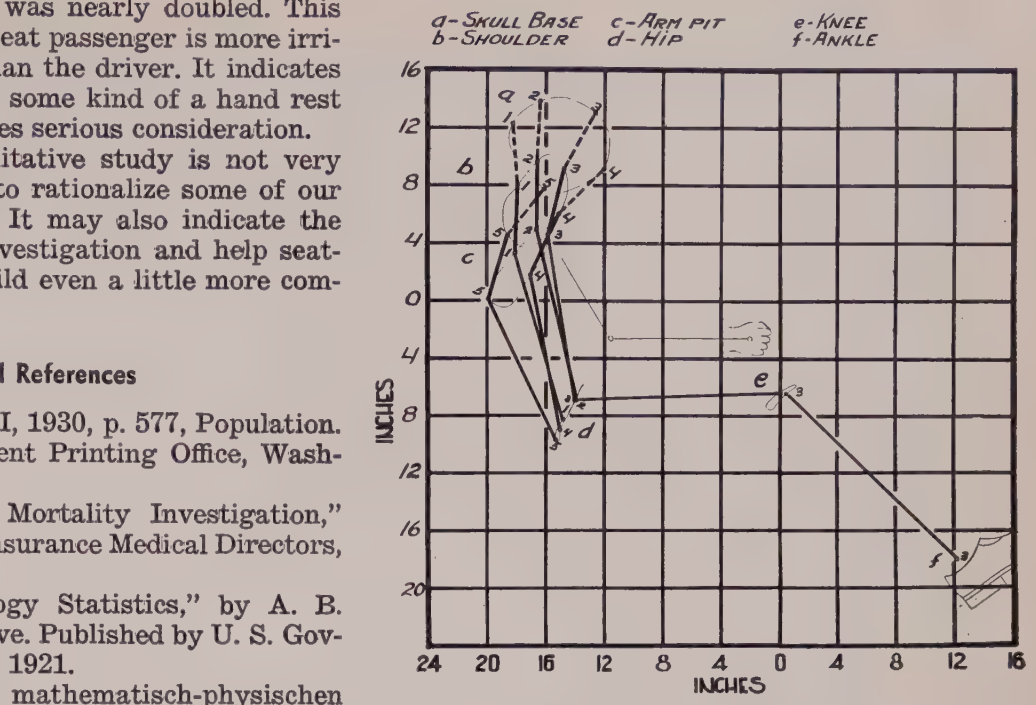

Fig. 5 – Movement of spine and thigh

Additional References

1. U. S. Census, Vol. II, 1930, p. 577, Population. Pub. by U. S. Government Printing Office, Washington.

2. "Medico-Actuarial Mortality Investigation," by Association of Life Insurance Medical Directors, New York, 1912-1914.

3. "Army Anthropology Statistics," by A. B. Davenport and A. G. Love. Published by U. S. Government Printing Office, 1921.

4. Abhandlungen der mathematisch-physischen classe der Königlichen Sachsischen Gesellschaft der Wissenschaften, Vol. 15, 1889, "Uber den Schwerpunkt des Menschlichen Körpers," by W. Braune and O. Fischer.

5. "Riding Qualities of Motor Vehicles," by R. W. Brown. SAE Transactions, Vol. 20, Part II, 1925, pp. 416-438.

6. "Preliminary Report on Fatigue Produced by Automobile Riding," by F. A. Moss. SAE Transactions, Vol. 24, 1929, pp. 15-20.

7. "Measurement of Comfort in Automobile Riding," by F. A. Moss. SAE Transactions, Vol. 25, 1930, pp. 7-12.

8. "Bodily Steadiness – A Riding Comfort Index," by F. A. Moss. SAE Transactions, Vol. 25, 1930, pp. 13-17.

9. "New Riding-Comfort Research Instruments and Wabblemeter Applications," by F. A. Moss. SAE Transactions, Vol. 27, (April) 1932, pp. 182-184.

10. Purdue University Engineering Bulletin, Research Series No. 44, Vol. 17, May, 1933, "Riding Comfort Analysis," by H. M. Jacklin and C. J. Liddell.

11. "Criteria Are Set for Riding Comfort, Research; New Instruments Made," by R. W. Brown and H. C. Dickinson. *SAE Journal,* Vol. 37, August, 1935, pp. 20-23.

12. "Automobile Seat Cushions and Riding Comfort," by Clark Tea. *SAE Journal,* Vol. 43, July, 1938, pp. 26-29.

13. "Seat Cushions and the Ride Problem," by C. R. Paton, E. C. Pickard, and V. H. Hoehn. SAE Transactions, Vol. 35, (July) 1940, pp. 273-283.

14. "Riding Comfort and Cushions," by W. E. Lay and L. C. Fisher. SAE Transactions, Vol. 35, (November) 1940, pp. 482-496.

15. "A Survey in Seating," instituted by Heywood-Wakefield Co. and conducted by E. A. Hooton and the Harvard University staff. Pub. by Harvard University, Cambridge, Mass., 1945.

16. "Ride and Vibration Data – Set of Reference Charts." Compiled by SAE Riding Comfort Committee. Pub. by SAE as SP-6, 1950.

17. "Mechanics of Human Muscles," by H. W. Haggard. *Mechanical Engineering,* Vol. 68, April, 1946, pp. 321-324.

18. "Machines to Fit People," by C. T. Morgan and J. W. Geohard. *Science Illustrated,* Vol. 2, November, 1947, pp. 42-45.

19. "Toward Better Cockpits – An Engineering Approach," by S. A. Hall. *Aviation,* Vol. 46, January, 1947, pp. 47-51.

20. "Psychology and Aircraft Design," by Paul M. Fitts. *Mechanical Engineering,* Vol. 69, February, 1947, pp. 135-141, 163.

21. "Greater Comfort for the Airline Passenger," by K. R. Jackman. *Aero Digest,* Vol. 57, October, 1948, pp. 62-63, 65-67, 119-122; November, 1948, pp. 42-46, 92.

22. "Walls of Jericho," by L. E. Muller. Presented at SAE National Passenger-Car and Production Meeting, Detroit, March 5, 1948.

23. "Vehicle Vibration Limits to Fit the Passenger," by R. N. Janeway. Presented at SAE National Passenger-Car and Production Meeting, Detroit, March 5, 1948.

24. "Handbook of Human Engineering Data," 2nd edition. Published by Tufts College, Institute of Applied Experimental Psychology.

Drive Line

THE truck engine of the future will be capable of producing more horsepower and will operate at higher rpm, according to Mr. Michell.

This means, he said, that new problems will arise in connection with the clutch, transmission, propeller shaft, and rear-axle design.

The author discusses some of these problems as far as each of the above parts of the drive line is concerned.

The Author

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RUMOR has it that the truck engine in several forthcoming editions will take the well-known V-8 configuration.

Following up the trend to over-square bore/stroke ratios, higher bmep's and piston speeds, one can arrive at startling engine speed and horsepower predictions, though the factors leading to these high speeds look quite feasible.

Let's take the bmep at governed speed, now around 100, and raise it to 110.

The bore/stroke ratio, already over-square, will go as high as 1.18 in one crystal ball.

The piston speed, from the forecast, can go up to 2600.

Combining these maximum factors into an eight, we have curve 196X in Fig. 1, with speeds about double those of current 6-cyl engines. Present fast sixes lie along curve 1954. More conservatively, an intermediate design objective curve 1957 might apply.

Horsepower at much higher speeds seems to be predicted for the clutch, transmission, propeller shaft, and rear axle, with new problems.

It looks as if the unit builder should check up on his designs and ratios.

Vehicle

To bring out, as nearly as we can foresee, the problems in the drive-line units, let us try to forecast a future highway combination with specifications as given in Table 1.

Using the conservative figures in the SAE "Truck-Ability Prediction Procedure," we find the horsepower required for various speeds as shown in Table 2.

Aiming at a safe 50-mph speed prediction, we need 190 hp, bare engine; using the 1.18 bore/stroke ratio and 110 bmep at governed speed, the engine would come out as shown in Table 3.

For some time to come, this engine will run at 3500 rpm governed, but, judging by past experience, there will be a continuous effort to bring up the speed and horsepower.

The 3500 speed brings in a lot of design problems and, as a design objective, we will set up 4000 rpm and 192 hp as our goals, so that we will have reasonable life from our tool and other investments, in drive-line units.

Clutch

Several factors might require checking or test in clutch design.

1. Safety: We can take 13,000-fpm peripheral speed as a safe speed for cast-iron pressure plates.

If a 14-in. clutch is selected for our pilot project, it will at 4000 rpm have a peripheral speed of 14,700 fpm, with more speed possible.

So we may have to use smaller, multiplate clutches or new materials.

2. Size: Most automotive clutches are the size they are because some smaller sizes would not live. The size clutch for our new engine may be decided the same way, but this is leaving engineering up to the field.

For consideration here, a series of starts by a professional is shown in Table 4 for an 8-cyl engine.

Much more racing and heating up are required in the 15.9 ratio than in the 53.7, in spite of lower final speed.

The driver normally starts this rig in the 31.0 ratio. Faster ratios produce jerk and possible stall, and slower ratios waste time, in shifting up after

for High-Speed Truck Engines

W. P. Michell, Spicer Mfg. Division of Dana Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1954.

start. This checks well with the conventional belief that 10% gradeability is necessary to start.

This medium-high-speed engine shows ability to lock up and pull loads at low speeds, so we can say, with a couple of *if's,* that, with the high-speed engine, with lower torque, the driver will start in a deeper ratio, which is favorable to the clutch, and the clutch size should be smaller, assuming the engine will lock up and pull away at speeds comparable to present practice. (See, however, the comments of Harold Nutt on this subject, as given in the Appendix.)

3. Inertia: Calculations show that about two-thirds of the inertia to be overcome by synchronizers (or by teeth in a poorly timed double-clutched shift) comes from the clutch disc.

If we maintain present clutch sizes, this inertia is, roughly, going to double, since it varies with the square of the speed. Then the clutch disc energy change for a given ratio change will double.

The clutch disc moment of inertia varies with the fourth power of the diameter, so that if a 14-in. clutch is now used, a 12-in. clutch on a new, faster engine would compensate for increased speed as regards load on the synchronizers, but we have only hopes that we can reduce clutch size, as yet. The transmission engineer will have to use larger synchronizers, or reduce the speed change between shifts, if clutch disc inertia is not reduced.

Propeller Shaft

The problem in propeller shafts, with their universal joints, can be simplified by stating that for a given torque capacity we must carry a lot more speed.

The setup chosen has a 192-hp engine governed at 4000 rpm, with a maximum torque of 300 lb-ft.

We can find catalog sizes of propeller shafts from several companies to handle this, but only provided overdrive transmissions are avoided. Tentatively, let's state that we cannot go much higher than 4000 in this size of shaft assembly, and use this limit in transmission and axle design.

Our 1957 engines, then, are taking propeller shafts up to the upper limit of speed. Will further increases in speed of engines force a complete re-

Table 1 – Specifications for Future Highway Combination – Tandem Axle Trailer

Gross Combination Weight, lb	55,000
Tires	11.00/20
Vehicle Width, ft	8
Vehicle Height, ft	10
Cruising Road Speed, mph	50
Maximum Road Speed, mph	60

Table 2 – Horsepower Required for Various Speeds

	Mph				
	40	45	50	55	60
Bare Engine Hp	130	157	190	225	268

Table 3 – Engine for Future Highway Combination

Displacement, cu in.	345
Bore, in.	4
Stroke, in.	3 27/64
Maximum torque, lb-ft	300
Horsepower at various governed speeds:	
Rpm = 3500, Piston Speed = 2000, Hp = 167	
Rpm = 4000, Piston Speed = 2280, Hp = 192	
Rpm = 4500, Piston Speed = 2560, Hp = 215	

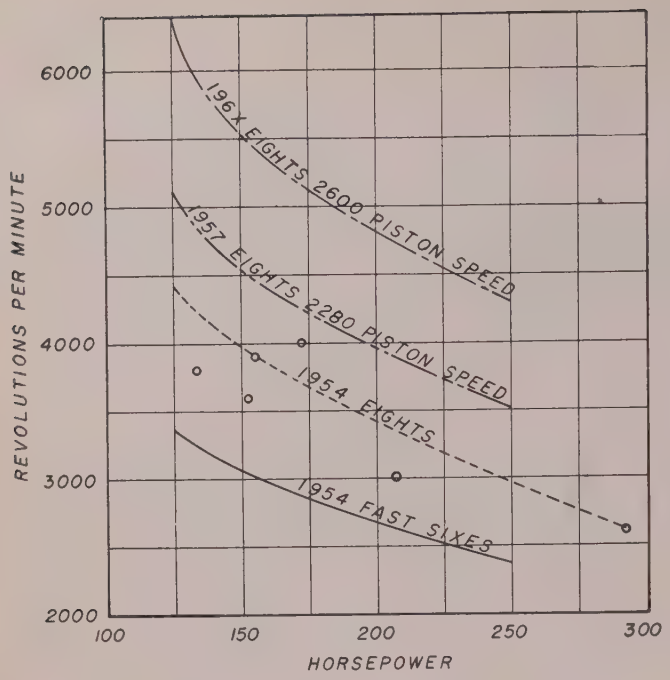

Fig. 1 – Engine forecast

design? There is an opening for a good prophecy here. Who can make it?

We can forecast the immediate moves in propeller shaft design as engine speeds increase:

a. Larger tubes, with lighter walls for higher critical speed.

b. Reduced permissible length, for higher critical speed.

c. Closer balance limits on all components.

d. Reduced permissible operating angles.

e. Reduced permissible out-of-parallel angle of end flanges, that is, engine, center bearing, and axle pinion must be held to closer parallelism.

f. More attention to mountings, particularly the midship or center bearing, which now sends considerable vibration into the frame.

An improved design of center bearing produces improvements as shown in Fig. 2. This bearing mount is designed so that critical speeds of mounting occur at a low road speed, and all operating speeds are above the critical.

After reviewing the temporary expedients, however, we may have to face up to the fact that the universal joints now standard were designed around torque at speeds that are now obsolescent, and the speeds have been and will continue to increase.

Development of shafts for speeds shown in the engine speed-horsepower graph (Fig. 1) will be a major enterprise, running into a lot of time.

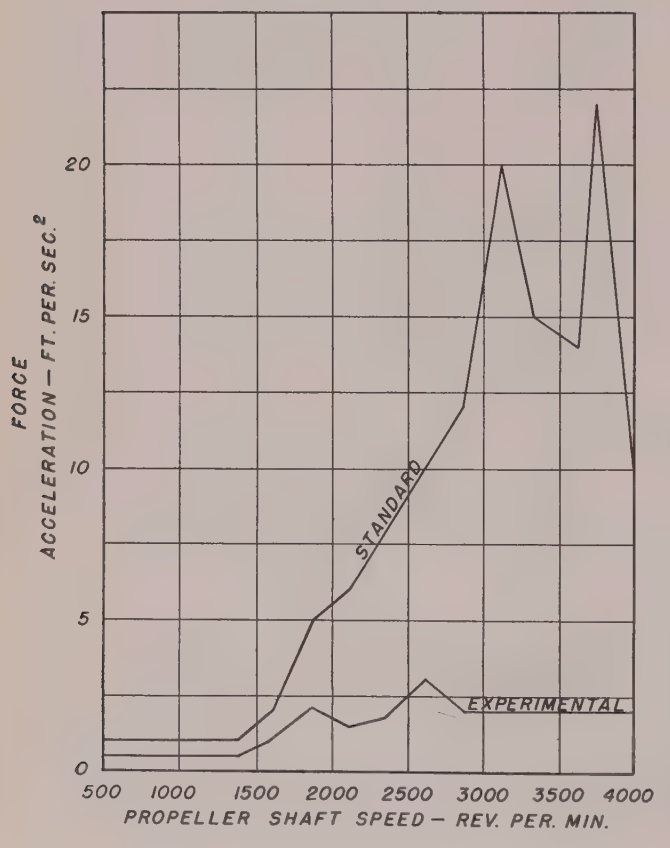

Fig. 2 – Vertical force transmitted through center bearing

Table 4 – Starts by Professional Driver

(Engine governed speed = 3200 rpm; hp = 143; torque = 284 lb ft; gross combination weight = 46,600 lb)

Starting Ratio Overall	Starting Gross Gradeability	Engine Speed at Lock-up	Road Speed at Lock-up
15.9	5.84	400	3.0
22.7	8.35	600	3.1
31.0[a]	11.4	1000	3.8
39.4	14.4	1200	3.6
53.7	19.7	1500	3.3

[a] Ratio preferred by drivers.

Axles

Table 5 shows the possible axle and transmission

Table 5 – Possible Axle and Transmission Combinations

	Cruise Speed, mph	Fast Speed, mph	Trans-mission High	Axle High	Axle Low	Prop Shaft Speed at 50 Mph	Prop Shaft Speed High
A. Overdrive Transmission, Cruise in Overdrive	50	None	.83	11.6	—	4800	4800
B. Overdrive Transmission, Cruise in 1.0 Ratio (Direct)	50	60	.83	9.65	—	4000	4800
C. Direct Transmission, Cruise in 1.0 Ratio (Direct)	50	None	1.0	9.65	—	4000	4000
D. Direct Transmission Cruise in 1.2 Ratio	50	60	1.0	8.05	—	3300	4800
E. Direct Transmission and 2-Speed Axle: Cruise in High Axle	50	None	1.0	9.65	12.0	4000	4000
F. Direct Transmission and 2-Speed Axle: Cruise in Low Axle	50	60	1.0	8.05	9.65	4000	4000

combinations on our guinea-pig tractor.

We can eliminate A as a useless speedup of transmission mainshaft, propeller shaft, and axle pinion.

B will have to be ruled out because of excessive propeller shaft speed in overdrive, until better propeller shafts become available. Having rejected A and B, we will not go into overdrive transmissions with 2-speed axles.

C will be acceptable in those fields where an overdrive speed is not now required.

D will be acceptable where an extra or overdrive speed is now insisted on. The job will run quietly with low losses in cruising, and without excessive noise at a speed of 60 mph.

E will interchange with C; the 2-speed ratio increases the overall gear low, and has a function in splitting the transmission ratios.

F is interchangeable, saleswise, with D; more care will have to be used on the drive line, since it is always running at 4000 rpm.

These combinations would all apply to future transmissions whether they be shiftable gear or semiautomatic, since we have the same options in the design of any transmission.

Since none of the axle ratios are less than 8 to 1, the present single-reduction axle, whose limit is 7 or 7.5, seems to be unusable with 8-cyl engines of the future.

While in D we show a compromise solution (cruising in 1.2 transmission reduction), economics will not permit any great transfer of reduction from the axle to the transmission; this adds to load on all drive parts – from transmission input to axle ring-gear. This is another way of saying that we can't get around high speeds by reducing them in the transmission. It seems, then, that larger axle ring-gears, permitting single-reduction ratios of 8 or 9 will be necessary. This means new housing tools.

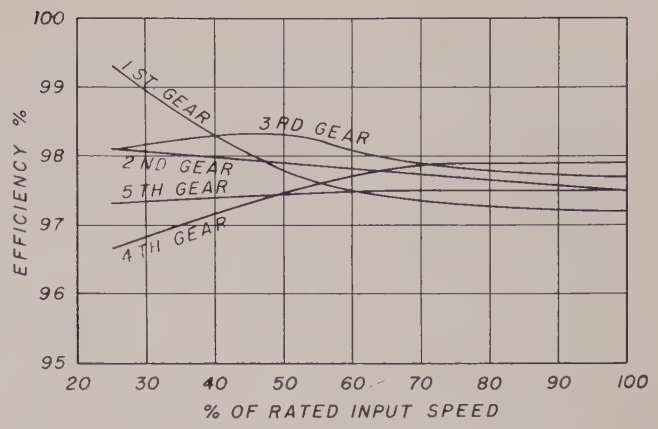

Fig. 3 – Transmission losses

Designwise, it is not believed that the differential and axle shafts will be affected by high-speed engines; their speed is limited by law and their load by the road grades and legal loads allowed.

Transmission Design

The problems in transmission design, as far as can now be seen, can be called off:

1. Size: If we conceive that new engines will simply run up the speed, gaining horsepower with present torque, we would be faced with higher cycles per mile on gear teeth and bearings, since the high speed goes through the transmission and as far as the axle. This would call for larger gears and bearings, particularly since dynamic forces on teeth increase with speed.

If, however, we run up the speed but reduce the torque to keep horsepower at present levels, we must balance the increased cycles per mile against

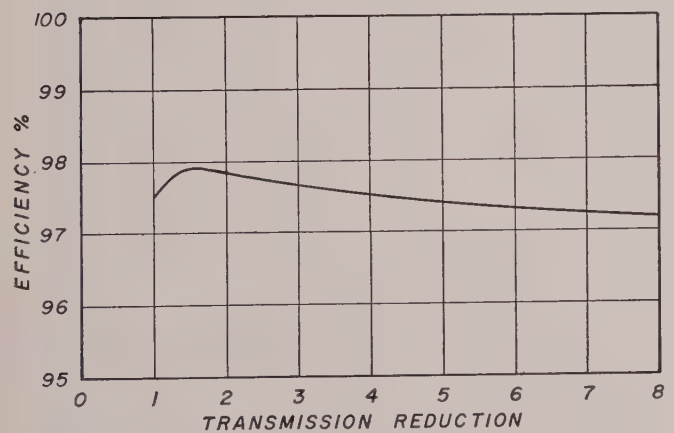

Fig. 4 – Transmission losses in ratios

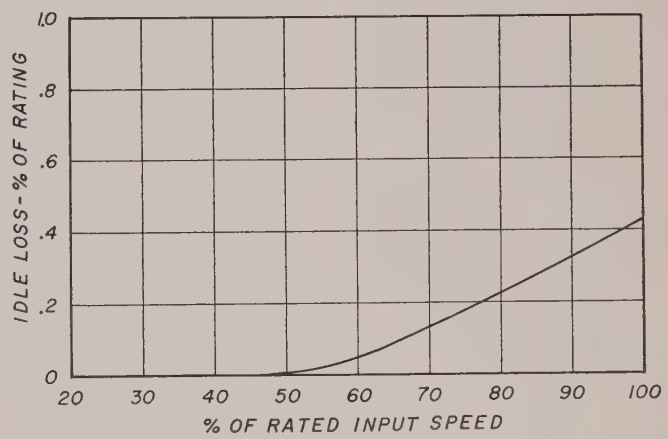

Fig. 5 – Transmission idle losses – loss versus input speed in high gear

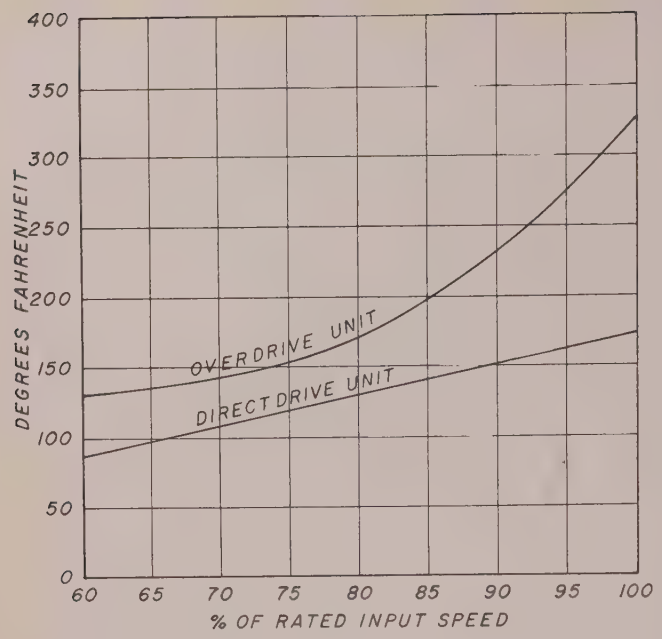

Fig. 6 – Stabilized temperature rise above ambient for two overfilled transmissions

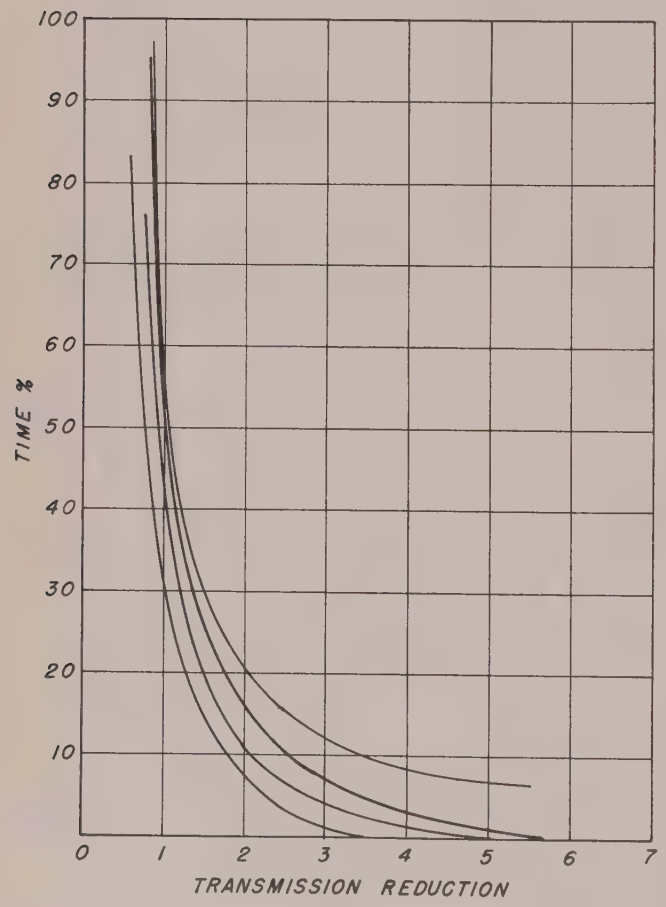

Fig. 7 – Sample road records – transmission reduction versus % time more reduction required

the greatly increased life of gears and bearings with lower loads. But again, we must consider higher dynamic tooth loads, offsetting some of the gain.

2. Noise, vibration, and unbalance: These will require design and manufacturing changes of some magnitude but can be taken care of.

3. Power losses: Will higher speeds result in serious power losses in transmission?

Fig. 3 shows efficiency in all gears, at full load, in a modern transmission for current high-speed engines. No serious loss of efficiency at higher speeds is indicated.

We may be interested in cruising in gears; Fig. 4 shows there is no appreciable loss in running through gear meshes as opposed to driving straight through. The lower viscous and churning losses in underdrive compensate for gear mesh loss.

A sensitive test, idle loss versus speed is shown in Fig. 5. While these losses are small, they increase with speed, and produce higher operating temperatures.

Higher operating temperatures, it is well known, result also from high oil level. Fig. 6 shows stabilized temperatures in two transmissions, slightly overfilled, at various operating speeds. Note the higher temperature of the overdrive unit.

The temperature rise of oil can be quite serious in splash-type lubrication, at high speeds.

From this set of results:

a. Higher speeds per se do not result in serious losses.

b. Geared cruising speeds are as efficient as direct drive.

c. With high speeds the oil level is critical. Since temperature, even at present, is a serious factor, we should design for low oil level or dry sump.

Note the views expressed in the Eaton and Backus comments given in the Appendix.

Ratio-Wise – In planning transmissions for new engines we have a good chance to review our old concepts.

The primary object of the transmission in a road vehicle can be stated in many ways but, for our present discussion, let's say that the prime function is to deliver the maximum possible percentage of available horsepower-hours to the axle over a variety of road conditions.

In setting up ratios, other considerations are:

a. Starting. As discussed under the "Clutch" section of this paper, the driver prefers about 10% net gradeability to start on level ground. Engineers have tried to add another 10% to this, so that the vehicle can start on 10% adverse grade, and climb 20%, once started.

Since some of the largest trains run with total net gradeability of only 13 or 14%, the above 20% is by no means universal.

b. Slow speed for maneuvering is an occasional requirement on trailer trains, but will not be considered further here, since it is a local or individual

rather than a general requirement. Engine speed range helps here.

c. Accelerating is quite similar to hill climbing, but with most of the gear time spent in the speeds near the top.

d. For down-hill running, some users will demand a gear or speed faster than the top normal speed attainable, on the theory that the driver will go down hill fast anyway, and he should have a gear which saves the engine from overspeeding and removes the temptation to coast down hill, a dangerous practice.

e. Some of the gear ratios permit using the engine as a brake on down-hill running, but no special attention is required here, since plenty of ratios are usually available.

It would seem best to study the horsepower delivery as affected by ratios, then decide on ratios that cover this and the other factors, in the usual compromise.

With our new engine, with its higher horsepower and speed, we will no doubt have new ratios, but the roads and the loads will not change much, so that a form of operation analysis that may help our ratio selection in the new transmission can be applied in steps as follows:

1. In various units on highway hauling, obtain, over an entire trip, the time in each gear, in percentage of total trip time.

2. Extend these figures to overall gear ratio including axle, and to net gradeability in the particular gear. This gradeability is figured from engine peak horsepower, which comes out lower than using peak torque (we have to use each gear ratio up to top speed on the grade). AV^2 losses can be neglected.

These figures then show a competent driver's analysis of the run, with the combination furnished him. He runs in a certain ratio because he cannot get into a higher one, profitably.

3. By starting with 100% time and subtracting successively the time in each gear, we can plot these figures as probability or expectancy for the run as to:

Per cent time versus transmission gear use.

Per cent time versus overall ratio use.

New gear ratios can be spotted on the ratio

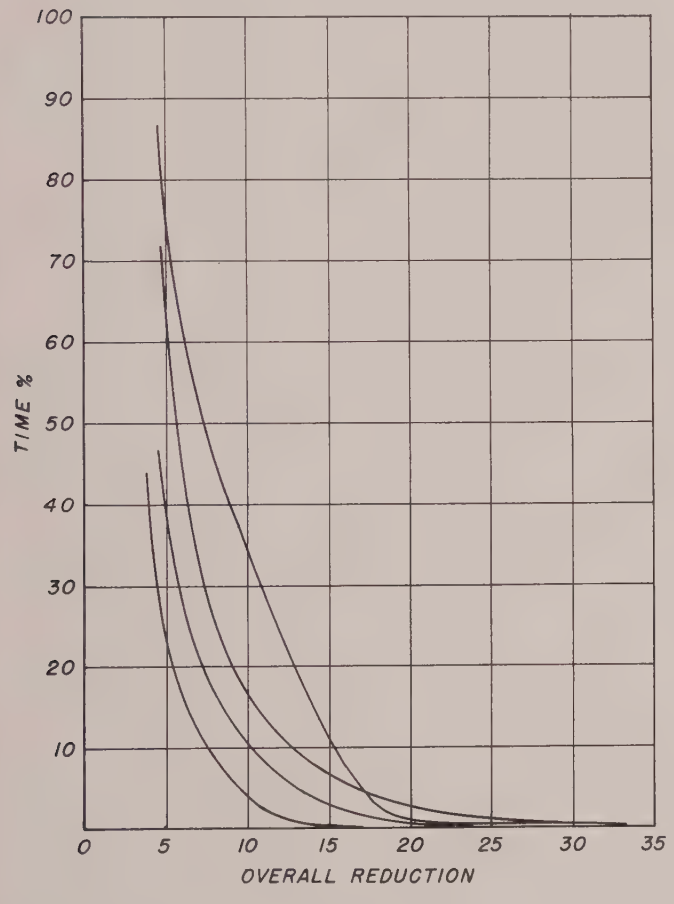

Fig. 8 – Sample road records – overall reduction versus % time more reduction required

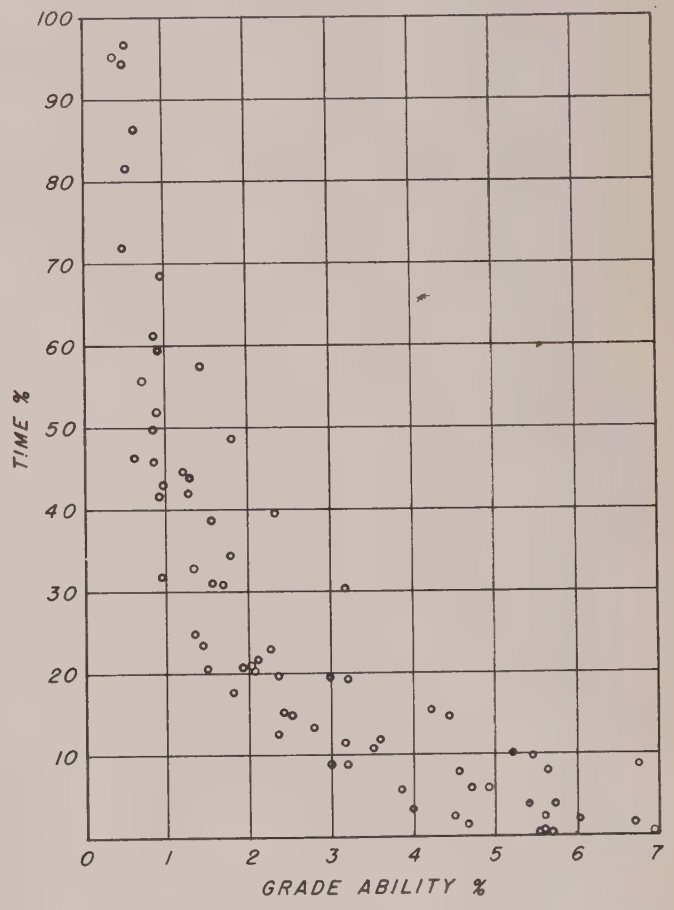

Fig. 9 – Road records – gradeability versus % time more gradeability required

curves, and time in each gear found by subtraction of its ordinate from that of the next higher ratio.

A number of runs with tractor-trailers have been clocked in various states and curves plotted, some of which are shown.

Fig. 7 shows several curves of time versus transmission gear ratio or reduction. Note the small use of gears slower than 2 to 1.

Fig. 8 shows the same data plotted for overall gear ratio.

Fig. 9 shows data for gradeability versus time for 10 runs in different areas and with different rigs. Gradeability can, of course, be translated to gear ratio for any tractor, given the data.

If our probability curve is a true one, we can change engine, transmission, axle ratio, or load on a run, and predict the time in each gear with the new combination, by a series of new steps on the old curve of gradeability.

If, further, we can accumulate enough data we can construct a general curve from which a transmission could be designed to best fit the average use. Fig. 10 shows such a curve, for rather severe requirements.

This type of record, plotted as a probability curve, could, aside from our present discussion, have some merit in analyzing engine conditions, comparing different engines, axle ratios, use of overdrive, and some other factors under proper test conditions, in most any automotive service.

Fig. 11 covers a high-speed 8-cyl engine and does not differ in shape from other curves as checked on this run.

Fig. 12 shows the idealized or general gradeability time curve, in which our new engine with an 8.05 axle ratio has been used to predict time in gear ratios, on log paper.

The rest of our ratio analyzing equipment will be:

Fig. 13: The engine curve showing horsepower against engine speed, both in percentage of governed maximum.

Fig. 14: The axle horsepower available from any gear step, using the assumption that the power output is the mean of the part of the engine curve used in that step. For example, a 1.50 gear step uses the engine speeds between 66.7% and 100%; and the power available through that step is 89.3% of maximum, using a common formula for curve mean. The gear step is always a quotient; divide the ratio in question into the next lower one. The

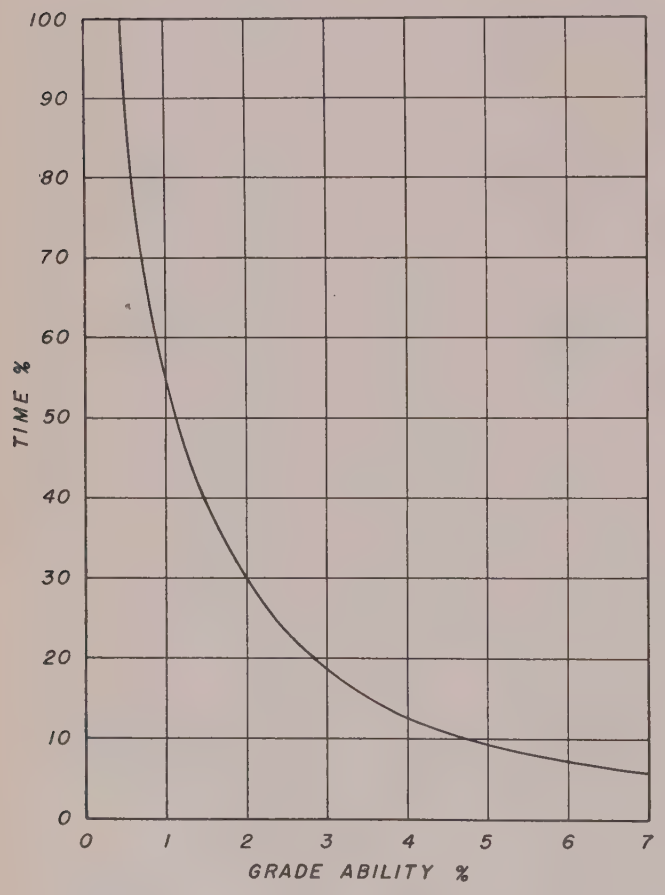

Fig. 10 – Gradeability for design use – gradeability versus % time more required – summary of more severe test runs

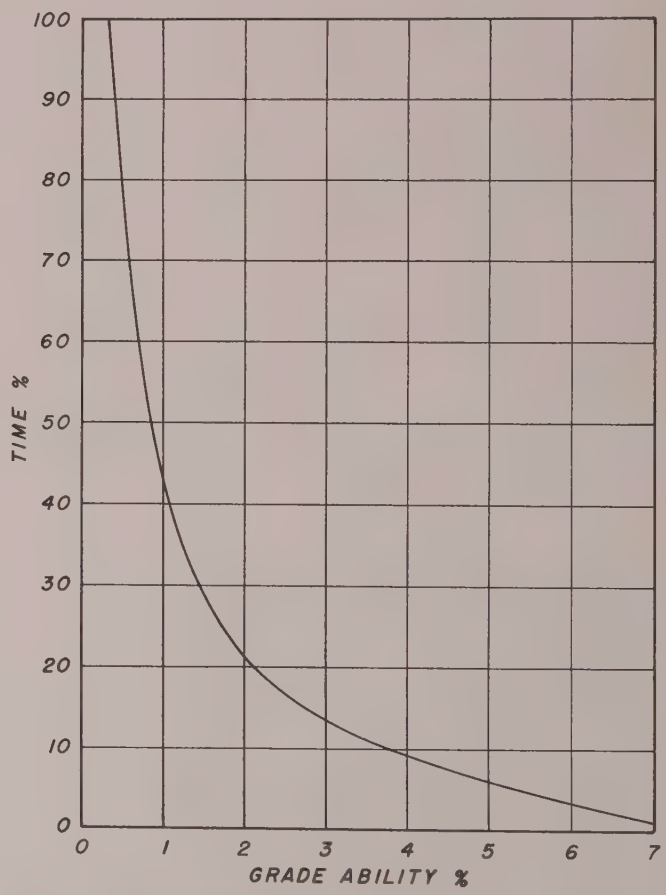

Fig. 11 – Road record – gradeability versus % time more gradeability required with 1953 high-speed 8-cyl engine

Table 6 – Data on 7-Speed Transmission with Uniform Step of 1.40

Speed	Ratio	% Time More Ratio Required	% Time in Ratio
7	1.00	70	30
6	1.40	40	30
5	1.96	24	16
4	2.75	13	11
3	3.85	7	6
2	5.39	3	4
1	7.55	1	2

engine runs between governed speed and the shift point to next lower step.

Now let us return to our idealized gradeability time curve. Take a number from 5 to 10 to represent the number of transmission speeds or ratios. Seven or eight speeds would be a fair guess, since we have increased the horsepower as against the common 5-speed transmission and 2-speed axle combination, which gives 10 speeds.

We now will try to set limits on our transmission ratios.

We found that a 9.65 axle would give us 50 mph, but undoubtedly a lot of our customers will want an optional speed faster than that, call it overdrive or what you will.

Since our transmission losses (see Fig. 4) are not seriously affected, and to avoid problems in driveline speed, we will work here with a transmission giving 60 mph in high, direct, and cruise in next to high.

This means our axle ratio will be 8.05, for 60 mph in high.

Our lowest ratio will be determined by starting; we will take a ratio giving 15.6% net gradeability

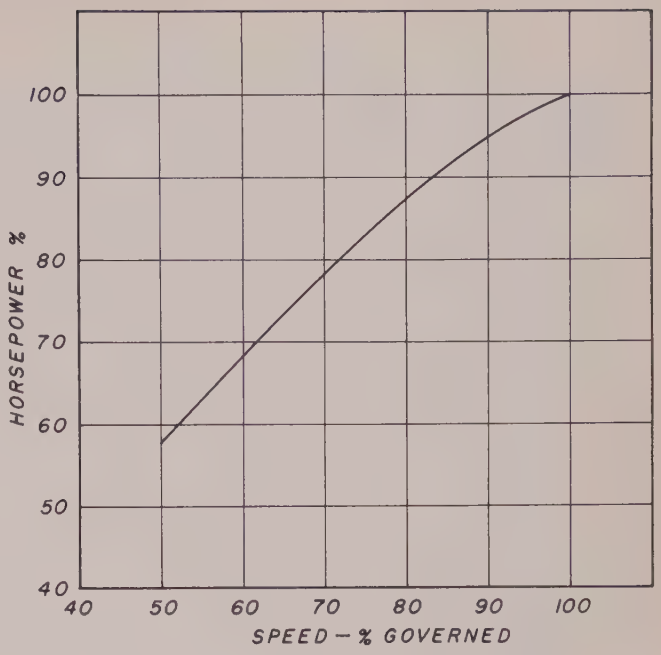

Fig. 13 – Engine horsepower versus speed

at engine peak torque. Transmission gear ratio for this is 7.55. This should start well on a 6% grade, and get away on a 10% grade.

We now have the problem of spacing our ratios from 7.55 to 1.00 in good steps, using seven speeds, as an example.

The simplest method is to divide the range into even geometrical steps, which is a widely accepted method. Our log paper graph makes this easy;

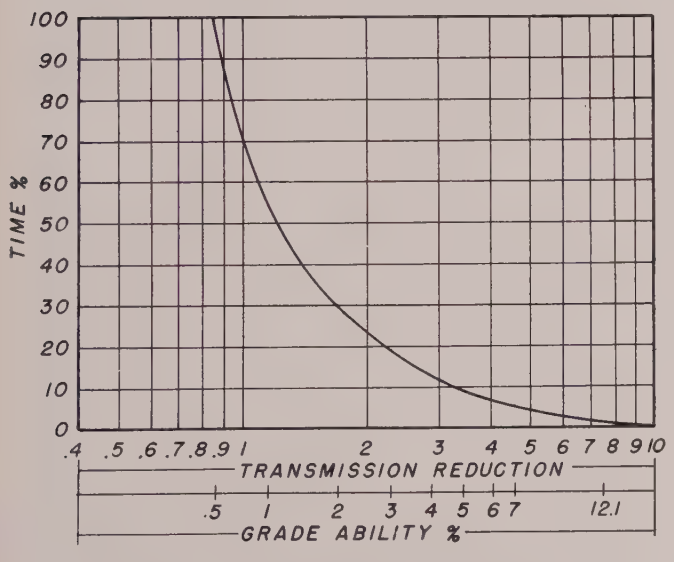

Fig. 12 – Gradeability for design use – same as Fig. 10 with transmission for 8.05 axle ratio and new engine

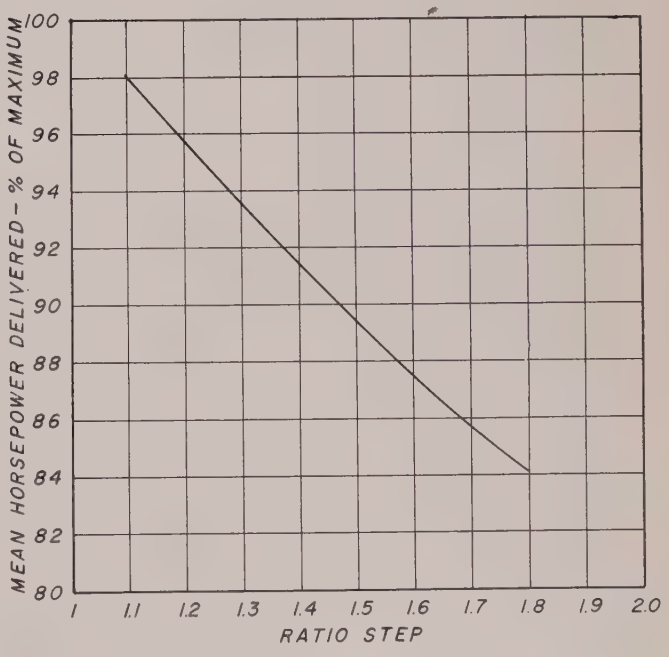

Fig. 14 – Mean horsepower delivered through various ratio steps

Table 7 – Data on 7-Speed Transmission with Equal Time in Each Ratio

Speed	Ratio	Down Step	% Time More Required	Minutes in Ratio per 100 Minutes	Mean Hp Delivered, %	Hp-Minutes Delivered
7	1.0	1.1	70.0	30.0	98.0	29.4
6	1.1	1.14	58.4	11.6	97.5	11.3
5	1.25	1.20	46.7	11.6	95.8	11.1
4	1.5	1.33	35.0	11.6	93.4	10.8
3	2.0	1.5	23.3	11.6	89.4	10.4
2	3.0	2.5	11.6	11.6	81.5	9.45
1	7.55	—	0	11.6	81.5	9.45

Data in first gear (seldom used at full throttle) must be extrapolated.

divide the horizontal space into six equal spaces.

This procedure results in a 7-speed transmission with a uniform step of 1.40, as shown in Table 6.

An 8-speed combination would give a step of 1.33.

Fig. 14 shows the mean horsepower delivered:

7 speed — 1.40 step — mean hp = 91.5% of maximum

8 speed — 1.33 step — mean hp = 92.8% of maximum

In either case, we must alter the step below high to get 50-mph cruising.

There has always been a demand for one or two closely stepped ratios near the high end, and the road record curves show the reason: very little time is spent in ratios deeper than 2 to 1.

If we could allot more gear steps in the high end, less in the low, the mean horsepower output should improve. A radical method would be to allot equal times to each ratio, by laying off equal spaces on the time scale (Fig. 12).

This, in a 7-speed transmission, gives data as shown in Table 7.

Adding the hp-minutes delivered and dividing by total time, we get over the whole range 92.5% mean horsepower delivery, which is about equal to the eight speed. On the less severe, more typical runs, this mean horsepower comes out around 95%, roughly equal to a 10-speed (with even steps of 1.25) gearbox for this range. The even step boxes show the same output in all services, while the gathered ratio box improves as we get away from severe grades, which penalize wide steps in lower gears.

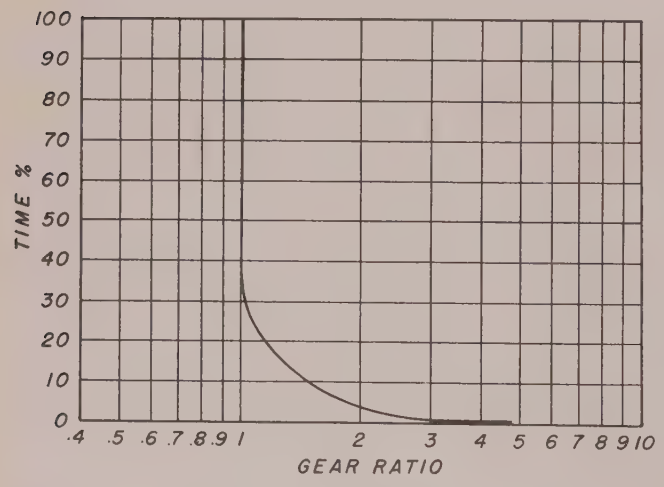

Fig. 15 – Use of gear ratios in accelerating tractor-trailer – % time more ratio required versus gear ratio

By transmission, in the foregoing, we mean any ratio changing means whether it be single gearbox, compound gear box, or 2-speed axle.

In practice the steps would have to be modified to suit the gear train, and to avoid the large step at the low end.

To accelerate a loaded tractor-trailer up to full speed usually requires four to five minutes; this cycle is repeated many times in a day's work. The time to 30 mph is one minute, more or less, so that any slowdown on the road brings in an accelerating cycle of probably three or more minutes.

Most of this time is spent in the higher gears; from Fig. 15, for a heavily loaded rig, it appears that the ratio below high is critical, since the two top gears do most of the work, and close steps show saving of time.

Judging by data, past practice, and new designs now in progress, we will have:

1. Not less than seven or eight usable speeds.

2. Ratio steps closed up or gathered near the high end, if shifting and gear pattern permit, or more steps provided for equal performance.

3. Transmission range reduced only if increased horsepower is available. The wider speed range of engine will not affect this transmission range.

If a torque converter or fluid drive is used, the gear steps must follow the same pattern, down to ratios that show little use on the road records.

In closing, the writer extends thanks to the many individuals in the industry who have contributed valuable advice and time, assisting in this job.

APPENDIX I

The following comments were received by the author in response to a request for information on the subject of this paper:

1. T. Backus, Fuller Mfg. Co., made the following comments:

It is our opinion that there is apt to be little, if any, effect on transmission size and size of parts, since horsepower, rather than torque, is the controlling factor.

There will unquestionably be an increase in noise, which will necessitate either quieter gearing or a new approach to the acoustics of the situation.

Lubrication will also present an increasing prob-

lem, which may have its effect not only on the quantity of oil used but on the means of lubrication and on the type of oil used.

Obviously, with higher speeds, gear balance will become more important.

The emergency brake is seldom used as such today and, unless it becomes a real emergency brake instead of just a parking brake, the only problem it will present is that we must watch its flywheel effect, with particular reference to balance.

Power losses will increase unless the solution to the lubrication problem can also be made to solve this problem. The same can be said of the effect of the heat on the transmission. In other words, lubrication, power losses, and heat are all tied in together.

2. Ernest E. Eaton, Clark Equipment Co., made the following comments:

We believe that high-speed engines will require re-evaluation of bearings and gears in transmissions. Also, items such as thrust washers and oil seals will undoubtedly require improvements. Without question, the high-speed engine will obsolete splash lubrication in the transmission and some form of spray or force-feed lubrication will be necessary, and certainly a higher degree of balance will have to be maintained on all rotating parts. We anticipate that with these refinements the gears and bearings themselves will carry higher loads with the same degree of satisfaction that they do today with the so-called slow-speed engine.

3. W. F. Scheel, Timken-Detroit Axle Co., made the following comments:

If engine speeds increase without a corresponding increase in propeller shaft speeds, very little change will have to be made in our present axle thinking. This means that we would have to forego the use of overdrive transmissions and resort to direct drive or even underdrive transmissions to keep vehicle speeds at 50 to 60 mph with increased engine rpm.

If drive-line speeds are increased correspondingly to engine rpm, axle ratios would have to be numerically higher. In this case, the basic axle design would most certainly be affected but the problem would not be insurmountable if the economics of the overall development justified new axle tooling.

4. Harold Nutt, Borg & Beck Division, Borg-Warner Corp., made the following comments:

The enclosed report (see Appendix II) outlines some approaches to the problem of clutch capacity.

It is to be noted that the expression covering the amount of energy required to start a vehicle does not include any factor relating to engine size. Also of particular significance is the fact that the overall ratio appears as a squared function. This means that the amount of work done by the clutch to start a vehicle varies inversely with the square of the overall ratio.

If we assume that the low-torque, high-speed engine is turning at the same rpm as the high-torque, low-speed engine during the clutch slipping period, and that the overall ratio has been adjusted to give the same torque at the rear wheels, the clutch will obviously have to absorb much less energy.

There is, however, some question as to the accuracy of the assumption regarding engine speed and overall ratio. It is our opinion that starting a vehicle with a low-torque engine and a high overall ratio will not be as satisfactory to the operator and will require more gear shifting than the higher-torque engine, which develops its maximum horsepower at a lower speed.

It has been standard practice in the clutch business to predicate clutch size on engine torque, with little attention being paid to horsepower. Vehicle weight and the type of service are, of course, important and that is the reason large clutches are often specified for severe service with comparatively small engines. It would be reasonable to specify a somewhat smaller clutch in a given vehicle, if the engine torque is reduced, provided you could be assured that the clutch would be engaged at the same engine rpm and in a gear ratio that would give the same tractive effort at the rear wheels.

On the other hand, we feel that this is a condition that could not be assured, and we would be inclined to maintain the present clutch sizes on any recommendations that we would make for the high-horsepower, high-speed, low-torque engines.

APPENDIX II

Determination of Clutch Capacity

by Harold Nutt, Borg & Beck Division, Borg-Warner Corp.

For automotive or other purposes, it is usually essential that the designer first establish the diameter of the clutch that is to be used behind the powerplant. In passenger-car service, this can be easily determined with a fair degree of accuracy from the formula:

$$D = \sqrt{\frac{T}{2.2}}$$

where:

T = Engine torque, ft-lb

D = Outside diameter of driven disc, in.

This formula is based on having the inside diameter of the facings about 0.6 of the outside diameter.

Smaller inside diameters are sometimes used but

the 0.6 rule gives about the maximum width of facing over which reasonably uniform distribution of load can be expected. In deriving the above formula, it is also assumed that about 0.45 sq in. of working facing area are provided per foot-pound of maximum engine torque.

From the above we have:

$$T = 2\frac{\frac{\pi}{4}\left(D^2 - (0.6D)^2\right)}{0.45} = 2.2D^2 \text{ (approximate)}$$

$$D = \sqrt{\frac{T}{2.2}}$$

A conventional single-plate automotive clutch with a driven disc having facings 10-in. outside diameter and 6-in. inside diameter will handle a passenger car with an engine having a maximum torque of 220 ft-lb and a loaded road weight of 4200 lb. Checking with the preceding formula:

$$D = \sqrt{\frac{220}{2.2}} = 10 \text{ in.}$$

The necessary total spring load may now be computed from the formula:

$$P = \frac{6T}{\mu R_f}$$

where:

T = Torque capacity, ft-lb
μ = Coefficient of friction
R_f = Mean effective radius, in.

For a proper margin of safety the value of μ may be taken as 0.22. At normal engaging temperatures, the actual value of the coefficient is usually about 0.30, but the coefficient can fall as low as 0.22 at the high temperatures that are encountered during periods of abusive slip. The mean radius may be taken as the arithmetic mean rather than the true moment radius, which would be at a distance from the center equal to:

$$R_f = \frac{2}{3}\left[\frac{R_1^3 - R_2^3}{R_1^2 - R_2^2}\right]$$

where:

R_1 = Outer diameter of facing
R_2 = Inner diameter of facing

Because the coefficient of friction varies widely, precise calculations of capacity are meaningless and the error introduced by using the arithmetic mean radius may be neglected.

For the example already mentioned, the required spring pressure would be:

$$P = \frac{6 \times 220}{0.22 \times 4} = 1500$$

where:

$$R_f = \frac{10 + 6}{4} = 4 \text{ in.}$$

Table 8 – Clutch Capacity Factors

Passenger Car	$T = 2.2D^2$	$D = \sqrt{\frac{T}{2.2}}$
Truck	$T = 1.7D^2$	$D = \sqrt{\frac{T}{1.7}}$
Lift Trucks and Other Very-Heavy-Duty Units	$T = 0.9D^2$	$D = \sqrt{\frac{T}{0.9}}$

The preceding computations are based on the assumption that all the spring pressure is applied to the driven disc. If opposing loads such as lever return springs are present, proper allowance must be made for their effect. The capacity factors for several types of service are shown in Table 8.

In addition to rating heavy-duty applications lower for a given diameter, it is customary to provide additional spring pressure as a protection against loss of load at the higher average operating temperatures that are encountered.

A single-plate clutch, if not too large in diameter for the available space, is usually to be preferred to a multiple-disc clutch of smaller diameter. The latter is more difficult to cool, with the result that distortion of the driven discs can make clean release a problem.

The diameter of a clutch is generally limited by the space available. This is definitely the case with motor vehicles, where the clutch must be located far enough above the ground for road clearance without raising too much of a hump in the floor boards. In applications where there is unlimited space, the maximum diameter permissible is usually controlled by the safe bursting strength of the pressure plate. A good grade of cast iron similar to that used for cylinder blocks or flywheels may be run up to a maximum peripheral speed of 13,000 fpm without danger of bursting.

Because of the uncertain character of cast iron, higher speeds, although possible, are not recommended. It is important that the design of the pressure plate be such that the cross-section is at no point seriously weakened by a reduction in area, or heavily overloaded locally by parasite metal, which adds stress but contributes nothing in the way of strength.

No material is quite as satisfactory as cast iron for a clutch friction surface. Steel should be avoided because scoring is much more likely to take place at the high surface temperatures that are frequently encountered in clutch operation. Where

steel must be used, scoring will be less severe if the carbon content is held above 0.30.

Although the foregoing roughly covers the method of computing clutch capacity, a more scientific approach would base clutch size on the work done per second per square inch of facing area for a given period of time. In other words, enough facing area must be present to absorb the energy of slip at a certain horsepower rate for a specified length of time without overheating. This statement assumes that pressure plates are of proper proportions to absorb the heat generated at the surface during the slipping cycle. Here, experience with the class of service involved is essential in the final determination of size and proportions of the assembly. While it is customary to rate clutches on the basis of engine torque, adjustment factors must be introduced to cover the widely varying types of service. Why this is so will be apparent from the following analysis.

To start a vehicle with a friction clutch, a certain amount of energy must be expended. If it is assumed that the throttle is wide open and the clutch engaged to a point where the engine is held at constant speed during the entire period of slip, the vehicle will accelerate at a constant rate. The clutch will slip until a vehicle speed is reached that corresponds to engine speed. During the period of slip, the amount of energy dissipated as heat is equal to one-half the total energy output of the engine. This is evident when one considers that all of the energy of the engine is turned into heat at initial engagement before the vehicle moves and none is wasted as heat after the vehicle has reached synchronous speed. In actual practice, some deviation from this condition may be expected because few operators are able to hold a clutch engaged at full throttle just the right amount to maintain a constant engine speed.

The above conditions could, however, be met with a centrifugal clutch, and it will be assumed for the purpose of establishing a standard of comparison that constant engine speed and torque are maintained during acceleration.

The other half of the output of the engine is transformed into kinetic energy of the vehicle. The energy distribution may be usefully expressed in terms of vehicle weight, gear ratio, and wheel size as shown in the following analysis:

Let:

m = Mass of vehicle

w = Weight of vehicle, lb

f = Force of acceleration (tractive effort), lb

a = Acceleration, ft per sec per sec

g = Acceleration of gravity, 32.2 ft per sec per sec

V = Velocity of vehicle when clutch slip ceases, fps

s = Revolutions per second of engine during period of slip

T = Engine torque, ft-lb

t = Time to bring vehicle in synchronous speed, sec

E = Total work delivered by engine in time t, ft-lb

E_c = Total work absorbed by clutch as heat, ft-lb

E_v = Total work absorbed by vehicles in the form of kinetic energy, ft-lb

r = Overall ratio in starting gear

H = Wheel rolling radius, ft

$$E = E_c + E_v$$

$$E_c = E - E_v$$

$$E = 2\pi Tst \text{ (force times distance)}$$

$$E_v = \frac{mV^2}{2} = \frac{wV^2}{2g}$$

$$V = \frac{2\pi sH}{r}$$

$$E_v = \frac{4\pi^2 s^2 H^2 w}{2gr^2} = \frac{2\pi^2 s^2 H^2 w}{gr^2}$$

$$t = \frac{V}{a} = \frac{2\pi sH}{ra}$$

$$f = \frac{wa}{g} \quad \text{also} \quad f = \frac{Tr}{H} \text{ (tractive effort)}$$

$$\frac{w}{g} a = \frac{Tr}{H} \quad \text{and} \quad a = \frac{Trg}{wH}$$

$$t = \frac{2\pi sH}{r} \times \frac{wH}{Trg} = \frac{2\pi sH^2 w}{Tr^2 g}$$

$$E_c = 2\pi Ts \left[\frac{2\pi sH^2 w}{Tr^2 g} \right] - \frac{2\pi^2 s^2 H^2 w}{gr^2}$$

$$E_c = \frac{2\pi^2 s^2 H^2 w}{gr^2}$$

$$E_c = \frac{0.613 s^2 H^2 w}{r^2} = E_v$$

From the preceding computations it is evident that the work absorbed by the clutch E_c is directly proportional to the weight of the vehicle, the squares of the engine speed and rolling radius of the rear wheels, but inversely proportional to the square of the overall gear ratio. Note that there is no factor related to engine size in the formula. It takes the same number of foot-pounds of energy to accelerate a mass whether heavy torque is applied for a short time or lighter torque for a longer time. The larger engine will get the vehicle under way more rapidly and rate of heat generation will be greater, but the total heat absorbed by the clutch will be the same. Nevertheless, engine size is an important factor since the rate at which heat is generated on the friction surface of the facings is a direct function of the engine output. The surface temperature developed during slip will be higher if the time of slip is reduced by the use of a larger engine. Wear of the facings will be accelerated because experience has shown that facings wear much more rapidly as the temperature of the surface increases.

It may be of interest to know what is considered a reasonable rate of heat absorption. Assume again a clutch with 10x6-in. facings. We know that this can be engaged at 220 ft-lb and 1400 rpm to start a passenger car without causing excessive surface

temperatures. The maximum horsepower developed during slip is:

$$\text{Hp} = \frac{220 \times 2\pi \times 1400}{33{,}000}$$

$$\text{Hp} = 58.7$$

The total working facing area is 100.5 sq in. Hence, maximum horsepower absorbed per square inch of facing area is 58.7/100.5, or 0.584, and the average hp per square inch facing area = 292. If the clutch is engaged at full throttle and at an engine speed corresponding to maximum engine output, the energy absorbed during acceleration may be over twice the energy absorbed at a lower speed, where maximum torque is developed. Such operation, while generating much more heat in the clutch, does not give best acceleration, but actually reduces tractive effort at the wheels. A clutch large enough for good durability will, fortunately, withstand occasional severe abuse of the type outlined above without serious consequences.

While clutch size is often based on engine torque alone, it is evident from the preceding calculations that a general formula should include gross vehicle weight, starting gear ratio, and rolling radius of the driving wheels.

Let us assume a passenger car with a loaded weight of 4200 lb, overall ratio in low gear of 11, rolling radius of 14 in. (1.167 ft), and 1400 rpm engine speed during the slipping period. Thus:

$$w = 4200 \text{ lb}$$

$$H = 1.167 \text{ ft}$$

$$H^2 = 1.36$$

$$r = 11$$

$$r^2 = 121$$

$$s = \frac{1400}{60} = 23.33 \text{ rps}$$

$$s^2 = 544$$

Then, from the equation for E_c:

$$E_c = \frac{0.613 \times 544 \times 1.36 \times 4200}{121} = 15{,}760$$

Thus, 15,760 ft-lb of energy are absorbed by the clutch.

Let E_s represent the total energy absorbed per square inch of facing area.

If the facings are 10-in. outside diameter by 6-in. inside diameter, the working area $A = 100.5$ sq in. and:

$$E_s = \frac{E_c}{A} = \frac{15{,}760}{100.5} = 157 \text{ ft-lb per sq in. work}$$

While it would appear from the preceding computation that clutch size can be based on certain known factors, a number of variables exist that must also be taken into consideration. Often the vehicle will not be started in the lowest gear ratio. This is particularly true in trucks with transmissions having four or more speeds. The driver usually selects the ratio that will permit him to get under way with reasonable acceleration and without too much risk of stalling the engine. This practice saves the driver some effort, but may greatly increase the rate of wear of the facings because the amount of energy absorbed by the clutch varies as the square of the overall ratio. Wear is also increased more than in direct proportion to the engine speed at which the clutch is slipped. High engaging speeds generate high temperatures on the friction surface and heat is the primary cause of rapid facing wear.

Frequency of operation increases wear approximately in direct proportion. If, however, clutch operation is spaced at very short intervals, as for example in commercial lift trucks, the average temperature tends to be higher than normal. Both conditions aggravate wear, and a much larger clutch is usually selected for this type of service. Increased size has the double advantage of more facing area and greater radiation surface.

It has been pointed out that the equation for E_c does not include engine torque. Nevertheless, engine size must be considered since the driver has the option of selecting any starting ratio available and may, therefore, increase the slipping time accordingly. It can be assumed that he will select a higher ratio with a large engine than with a small engine because he will have sufficient torque to get the vehicle going in a reasonable length of time. It is, therefore, probable that the actual energy absorbed by the clutch is a direct function of engine size as well as vehicle weight. However, the overall ratio and the wheel radius should appear as squared functions.

If we call the work factor K, we can then write the expression:

$$K = \frac{TwH^2}{Ar^2}$$

A study of current practice indicates that the value of K is about 100. Hence:

$$A = \frac{TwH^2}{100r^2}$$

where A is the total working surface of the facing in square inches. Using the example previously selected:

$$A = \frac{220 \times 4200 \times 1.167^2}{100 \times 11^2} = 104 \text{ sq in. (approx.)}$$

This formula is offered as a check on facing area where there is no background of experience to guide in the determination of use. It will be found, as always, that practice is a compromise and rather wide variations are the rule rather than the exception. Availability, reduction in the number of sizes to be carried in production, and the specifications of competitive equipment are but a few of the many factors involved.

Ordinarily, a clutch somewhat undersize will give moderately good results except in a small per-

centage of vehicles, which are subject to severe service. An oversize clutch is desirable where long life between facing replacements is important. Its drawbacks are additional original cost, greater weight, and the extra work required of the synchronizers when changing gears in the transmission.

Although the computations presented in this article may serve as a useful guide in the study of clutch installations, the designer should always seek the engineering advice of the clutch manufacturer, whose knowledge of his product usually covers a wide field of applications.

DISCUSSION

Considers Use of Double Reduction

– Leo A. Bixby
Bixby Management & Engineering Counsel

APPARENTLY the trend of the truck industry is to increase the rpm of engines, with a proportional increase in hp. Therefore, this, in all probability, will bring about changes in the drive line of the vehicle transmission, propeller shafts, and axles. However, many of us have the problem today of greater hp engines being used with very little increase of speed, in the same truck and with the same load conditions as with the past lower hp engines. In theory, therefore, we would not substantially increase the speed of the vehicle over what it is today, but in actual operation we will be maintaining a higher sustained speed, so it gets back relatively to the point of doing more work through the drive-line units to overcome the frictional hp loss, rolling resistance, and wind resistance, with less time in the miles traveled. Many trucks on the road today cannot operate on the level with the rated gross combination weight at a legal speed. Therefore, with these engines with high hp we will be spending less time in the lower ratios and moving that saved time into the upper ranges of the transmission, although the actual time spent in low speeds of the transmission will be less, the life hours might be decreased, due to greater hp and speed of the engine. The higher speeds in the transmission, however, will be called on to do more work at greater velocities, so it can very well be that those gears would have to be given attention to give satisfactory life. In operating at these higher speeds the gears have not only increased load but dynamic forces become greater, which will dictate better design and manufacture of the component parts of the transmission. However, with the high numerical ratios that will no doubt be in the axle with these high-speed engines, some of the maximum ratios we have used in the past may well give way to higher ratios with closer spacing, which will offer several methods of approach to help solve the transmission problem and use the increased rpm of the engines and operate high up in the hp range.

When trying to evaluate the axle under these conditions, we find that it, unlike the transmission, must deliver all the hp transmitted to it through the pinion at all times, while the transmission delivers a large per cent of operating hp in direct where the geared speeds are in neutral under no load other than dynamic forces. This analysis is based on not using overdrive transmissions with these high-speed engines, due to the excessive propeller shaft speeds, which would necessitate even lower axle reductions. Even with direct-drive transmissions it still will be necessary to have high numerical ratios in a single-reduction axle, which will mean fewer teeth in the pinion or substantially larger ring-gears, with resulting larger banjo diameters in the axle housings. Therefore, economy would dictate the tendency to use smaller pinions. These smaller pinions will have a decreased velocity factor, but this probably will be offset by the increase in speed and hp transmitted. This will accelerate wiping action with fewer teeth to take it and also the dynamic forces from the drive line will be greater on the pinion at these high speeds.

In view of this need to have high numerical ratios in the axles, we could give consideration to some form of double reduction, such as a final spur- or helical-gear combination, or a planetary means could be used at the differential or in the wheel ends. In that case the axle shaft life would be further enhanced. All of this will lend itself to using a combination of hypoids or spirals to the best advantage. It would also lend itself to using the smallest diameter banjo in the axle housings.

The obvious result with these engines of higher speed and hp is that it will make it possible to sustain higher mph of the vehicle. Therefore, the net answer is that the drive-line units will be doing more work in less time per mile traveled. The average operator is going to continue to want satisfactory life for his truck on the mileage basis, and it is going to be difficult, in most cases, to depart from that yardstick. Therefore, to maintain that standard of performance it will be necessary to continue the research for better steels, heat-treatments, manufacturing processes, lubricating oils, and design to meet these requirements.

Discusses Modifications for Some of Author's Conclusions

– A. F. Stamm
Chrysler Corp.

CERTAIN aspects of the discussion are based upon the power requirements calculated according to the conservative SAE Truck-Ability Prediction Procedure for a gross combination weight of 55,000 lb. While it is agreed that the SAE method tends to overestimate the power requirements, the author arrives at a need for 190 hp at 50 mph where the procedure, if applied as intended, would come out at about 169 hp. This discrepancy seems to come about by applying the friction horsepower equation to the total combination weight, where it is specifically limited to the rated capacity of the power unit only. The gross weight to apply is that of the tractor only. We have, for our use, modified the SAE values to get better agreement with our vehicles and would

Table A – Modified Values for Horsepower

	40 Mph	45 Mph	50 Mph	55 Mph	60 Mph
Hp (given in paper)	130	157	190	225	265
Hp (modified SAE)	85	100	122	145	172
Hp (SAE)	...	...	169	...	...

offer the comparison given in Table A. As another example, we can give the performance of a combination at 60,000 lb as 53 mph at 152 net hp. The vehicle cross-section was, in this case, 8 x 12 ft.

In regard to the clutch, it is perfectly true that the engine speed and the resulting peripheral pressure-plate velocity place definite limits on clutch diameter. At present, the clutch disc appears to be at least as good as the pressure plate, but, if higher speeds can be assumed to include a definite possibility of increased temperatures at the operating surfaces, it may well be that considerations of the facing material and disc configuration will become more critical. Furthermore, a very important point is brought out in calling attention to the relation between transmission shifting and clutch inertia. Higher speeds are going to be tougher on synchronizers.

For some time now, the propeller shaft has been on the verge of trouble. Even with some engines in current use, the problem of providing safe shafts for overdrive transmissions is not easily solved. Thin wall tubes are featured today but are of only very small help in raising the critical speed. Larger diameter tubes are very difficult to handle and do not appear to be a good answer for very much higher speed. As present limits of length and diameter are utilized, the difficulty and importance of close balance control cannot be stressed too severely. The increased magnitude of disturbing forces and the multiple modes of critical response, which are a penalty of high speed, dictate that joint angularity must approach zero and flange alignment must be nearly perfect. An ideal multiple shaft installation should produce very little reaction at the center bearings, but, nevertheless, bearing mountings of low vibration transmission characteristics, which are free from resonant periods within the operating speed range, are important today and will be even more difficult to design for still higher speed.

With a crankshaft speed of only 4000 rpm, which is scarcely beyond the present range, the selection of a suitable combination of driving ratios for axle and transmission becomes a difficult compromise for the present drive-line arrangement. As the number of engine cylinders increases and the stroke becomes less, the powerplant more closely approaches the performance of a turbine, which is characterized by high speed and low torque. Vehicle size and the performance requirements determine the horsepower needed. If a suitable turbine were available today, it would almost certainly be equipped with a reduction gear. It also appears to be a logical prediction to conclude that reciprocating engines for heavy-duty trucks that operate and peak at over 5000 rpm will also need reduction boxes. Contrary to the author's contention, the only way to get around extremely high speeds at the power source is to reduce them before they get to the clutch. With this approach the drive-line problems of the clutch, transmission, propeller shaft, and rear axle would remain about as they are today and capacity would increase only as operators insist upon greater performance, meaning more horsepower. Our present methods and practices can readily be extended to take into account more power but run into nearly insurmountable obstacles if the ceiling is raised on input rpm.

Whether our source of power operates at 3000 or 30,000 rpm, the road wheels will still be geared to cover a range of from about 0 to 60 mph. The problem remains one of matching the source to the load over the necessary speed range.

That an initial reduction in the powerplant is practical follows directly from the author's own data on gear efficiency and indirectly from cautions about the effect of high speed on safe lubricant levels, and increased operating temperature. The most feasible place to provide the special lubrication and cooling necessary for high-speed gearing is in direct connection with the engine, its oiling, and its cooling system.

A pattern of truck transmission ratios related to per cent of operating time engaged is essentially a record of the tractive ability demanded within a range of compromise that the highway transportation industry currently accepts. There is nothing inherent in the theory of the art that prohibits the possibility of realizing an ultimate performance potential such as is now being approached with certain passenger cars, meaning the ability to reach impending sliding of the driving wheels over a significant portion of the speed range. Truck transmission and axle ratios are at their present values because of the power developed by current engines and the gross weight of our highway combinations. At this moment it appears that the combination weights are not going to change much for some time; but, on the other hand, engine power is increasing. Little need for increasing the vehicle ability at the lowest end of the speed range can be justified, and it is also improbable that the upper speed limit at which trucks will be permitted to operate will be much different than obtainable with equipment now being built. Increases in power will be directed toward increasing vehicle flexibility, particularly in getting better sustained speeds on highway grades and better acceleration in traffic conditions. Within the limits of operation already established, trucks with more power will probably need modifications in transmissions. It seems probable that some form of hydraulic drive will gain greater use and only part of the ratio changing will be accomplished through gearing. With sufficient power available, the use of large reductions in low will be less justified and with the wide speed range of high-speed engines, it may well become practical to decrease the number of gear steps. No transmission or torque converter will ever replace the engine, and their importance as matching devices is inversely related to amount of power available in respect to loads handled and vehicle speed required.

Author's Closure To Discussion

MR. Stamm differs from the writer only in terms on the 190-hr engine to pull 55,000 lb at 50 mph. He gets 169 flywheel hp, whereas my figure was 190, bare engine. These two figures are quite in line.

The proposed modified SAE calculations are quite urgently needed, since common experience shows the present factors are too conservative.

On the subject of reducing the drive line speed, either ahead of the clutch or in the transmission, it must be agreed that such a step may become mandatory if engines go to the ultimate speeds predicted. Avoidance of overdrive is really a move in that direction. Further moves will, doubtless, be dictated by the degree of speed-up of engines and by economics.

Higher horsepowers, now or in the future, can simplify the transmission. Power/weight ratios in trucks can approach those of passenger cars, with resultant elimination of multiple gear steps.

Most owners will, however, still be conscious of fuel costs. This will place a limit on any extreme increase in horsepower in highway engines.

Turbojet-Engine Design Problems for Supersonic Flight

Joseph S. Alford and Earl L. Auyer,
Aircraft Gas Turbine Division, General Electric Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1954.

TODAY'S production turbojet engine was designed for flights in the subsonic speed range. Recently, with structural modifications, these subsonic engines have proved to be a satisfactory powerplant for an aircraft designed for speeds between Mach 1 and Mach 2, the low supersonic speed range.

In the years to come, flights above Mach 2 will become more important. Mach 2 flight requires some very stringent design requirements for both powerplant and aircraft. Let's discuss a few of the desirable features and more important problems involved in turbojet engines that are designed to operate a substantial part of their mission at Mach numbers between 2 and 3.

SOME of the aerodynamic and mechanical problems of jet engines designed for supersonic flight speeds are discussed in this paper.

The aerodynamic problems considered include the required range of operation of the compressor, the thermal efficiency of the cycle, the inlet-engine airflow match, and jet nozzle design. Structural difficulties due to high operating pressures and temperatures and the bearing and lube problems arising from high temperatures are also presented.

The Authors

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First, we will consider some of the general characteristics of an engine which will meet such speed requirements. In the supersonic speed ranges, the turbojet engine is a very efficient engine, contrary to general opinion. Flying supersonically, the turbojet has thermal efficiencies equal to or better than the maximum obtained with diesel engines or central station steam power systems. But it must be remembered that the rate of fuel consumption is high due to the large power requirements needed for flying at high speeds.

Fig. 1 shows the thermal efficiency at Mach 2 of a turbojet engine for various compressor ratios with temperatures at 1600 F, 1600 F with afterburner, and 2500 F. Thermal efficiency is here defined as the ratio of the energy output for overcoming the airplane drag divided by the thermal energy input of the fuel used.

As can be seen, the thermal efficiencies are very high and the curves quite flat, which suggests that moderate pressure ratios and high combustion temperatures will probably give the most practical engine.

At Mach 2, specific thrust, which is thrust per pound of airflow, must definitely be high in order to keep the engine size and weight to a minimum. Fig. 2 shows the values of specific thrust that can be obtained at Mach 2, and it can be seen that high turbine inlet temperatures or exhaust reheat with lower turbine inlet temperatures must be used.

The Mach 2 curves change drastically when Mach 3 flight speed is required. In Fig. 3, note that the high-temperature engine (2500 F) or the exhaust-reheat engine will give thermal efficiencies of from 35% to over 40% with compressor ratios as low as 3 or 4. Incidentally, these pressure ratios refer to

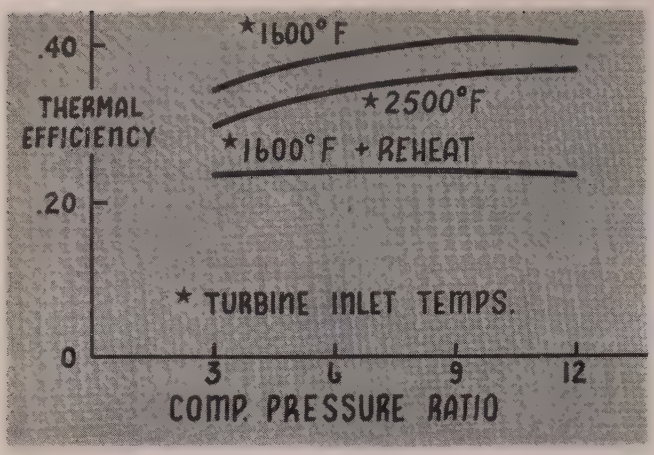

Fig. 1 – Thermal efficiency of a turbojet engine at Mach 2

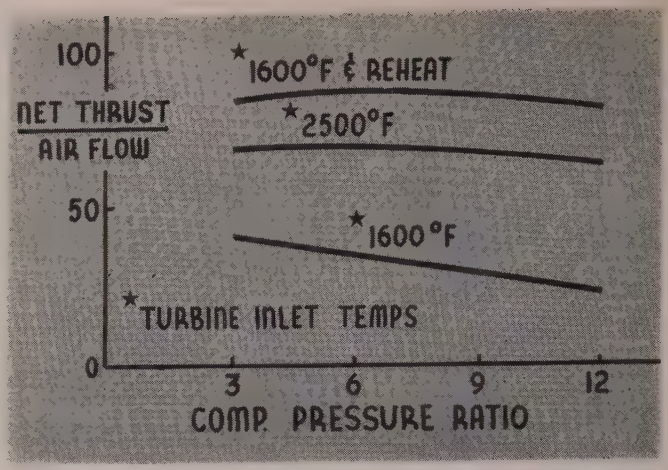

Fig. 2 – Specific thrust obtained at Mach 2

the actual pressure ratios at specific flight speeds.

With specific thrust at Mach 3, note the great change in the curves from the Mach 2 figures. In Fig. 4, the turbojet with reheat has high specific thrust but somewhat lower efficiency than the engine with high inlet temperature without reheat. Hence, the optimum high Mach number engine must have a moderately low pressure ratio and a high combustion temperature either ahead of the turbine or in the afterburner. Another inherent problem in engine design of supersonic engines is temperature. The energy put into the inlet air increases the temperature level at the compressor inlet and throughout the compressor as flight speed increases. Fig. 5 shows the compressor air temperatures plotted as the ordinate versus the Mach number as the abscissa. Numbers on the curves are the compressor pressure ratios when inlet conditions correspond to standard sea-level static atmosphere.

Calculations assumed constancy of the actual engine speed, turbine inlet temperature, and the turbine diaphragm flow area. The exact shape of the curves will vary with engine design.

High engine temperatures – both high ram air and compressor section temperatures – are the result of high speed. At Mach 3, compressor inlet or ram air temperatures reach 600 F, while compressor discharge temperatures reach approximately 1000 F. It must be noted that temperatures aft of the compressor section will not change appreciably at higher speeds unless new metallurgical or design developments permit raising the turbine inlet temperatures.

The temperature problem is a major one. While a great deal of talk has been aimed at the problems of aircraft skin temperature at higher supersonic speeds, the troubles of high temperatures in the engine are none the less important. Fig. 6 shows the expected temperatures at different areas of the engine. And all temperature problems are not

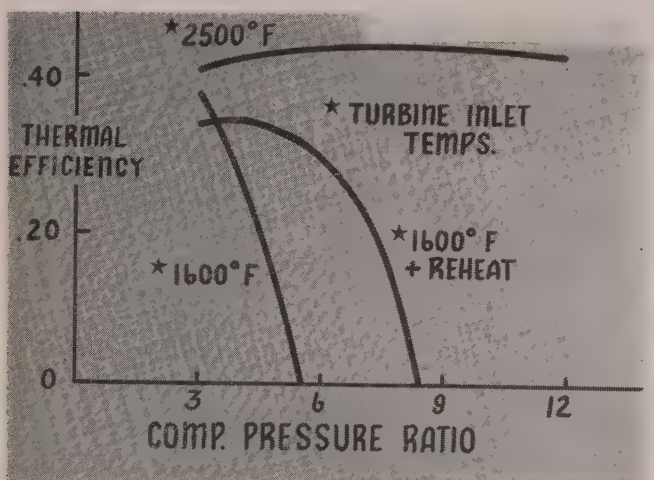

Fig. 3 – Thermal efficiency of a turbojet engine at Mach 3

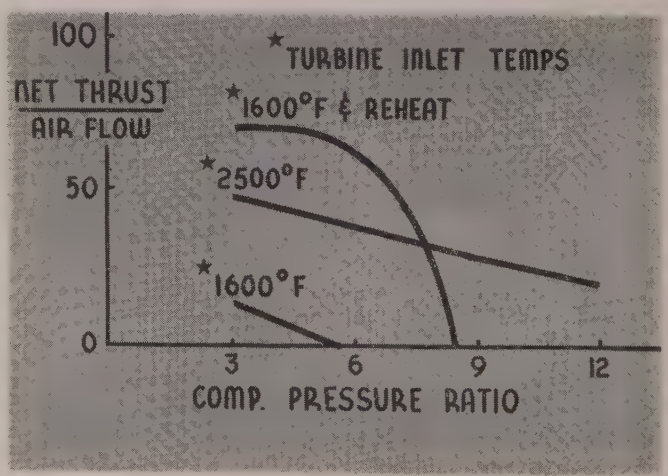

Fig. 4 – Specific thrust obtained at Mach 3

metallurgical. Visualize an engine with today's configurations running at full power in an oven heated to 600 F at a pressure of three atmospheres. At the present, it does not seem feasible to refrigerate enough cooling air to surround the engine with a cool atmosphere.

These high temperatures at higher supersonic speeds cause an increased fire hazard from fuel and lube oil. For example, the temperature of the high-pressure end of the compressor casing is above the ignition points of most hydrocarbon fuels and lubricants. The engine firewall must now become a duct to bring the fuel and lube lines to the engine. Hydraulic mechanisms may not be feasible unless cooled. This may lead to the development of more pneumatic systems. Electrical equipment will need further special attention. It may be desirable to remove the accessories from the vicinity of the engine and locate them in a separate insulated and cooled accessory compartment. All of the engine bearings are now hot-end bearings instead of the turbine end only as before. Refrigeration or cooling of the lube oil will be required. Development of higher-temperature lube oils will ease the bearing and cooling problems somewhat.

Higher temperatures at supersonic speed in turn affect pressure ratio. Maximum ratios diminish as speed increases because of higher inlet temperatures. Fig. 7 shows the approximate change in compressor pressure ratios with the increases in speed. It must be noted that the numbers on the curves are pressure ratios when inlet conditions correspond to standard sea-level static atmosphere. And again, calculations assumed constancy of actual engine speed, turbine inlet temperature, and turbine diaphragm flow area.

Thus we can see that to have an engine pressure ratio of 3 or 4 at higher supersonic speeds, the engine will have to be designed for a sea-level pressure ratio of approximately 9. For higher supersonic flight, pressure ratios will have to be increased throughout the engine, not only in the compressor section. Fig. 8 shows the absolute pressure in sea-level atmospheres versus Mach number. Calculations have been made for 40,000-ft altitude.

It will be noted that the compressor inlet pressure increases more rapidly than does the compressor discharge pressure. For example, at Mach 2.5, the compressor inlet pressure is 2.5 sea-level atmospheres. However, the compressor discharge pressure increases only 10-15% over its sea-level static pressure ratio of 9 or 12 at maximum rpm.

The most severe stress in wall casings is a cir-

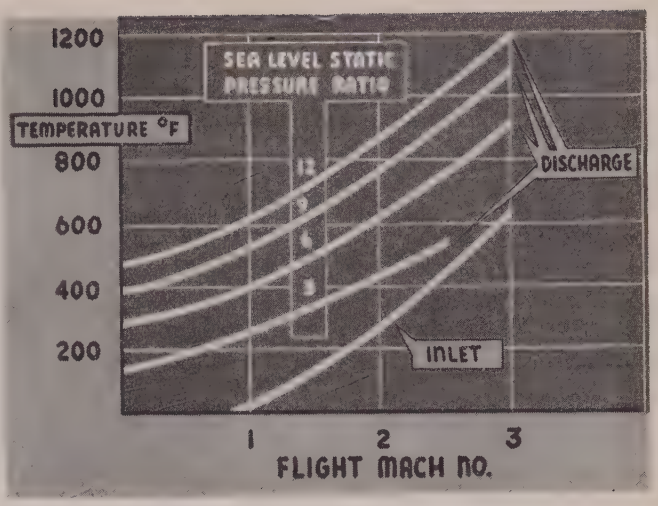

Fig. 5 – Compressor air temperatures versus Mach number

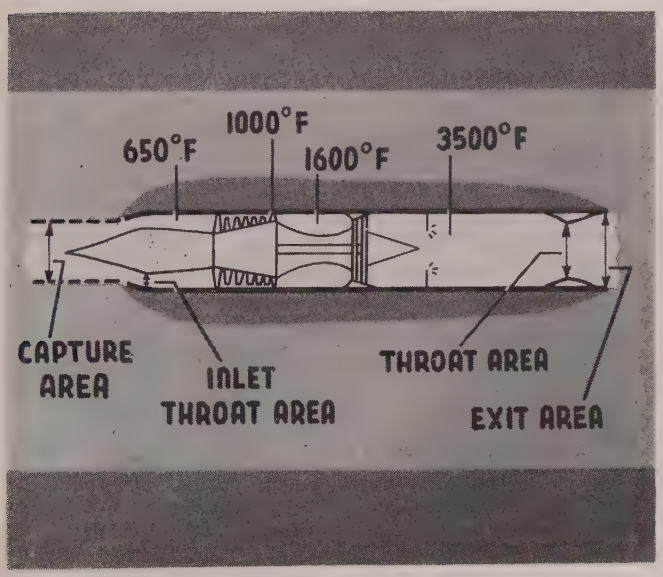

Fig. 6 – Temperatures encountered at different areas of turbojet engine

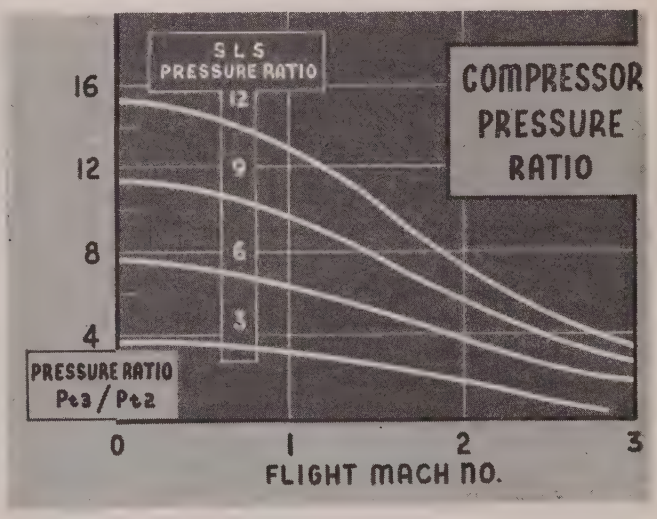

Fig. 7 – Compressor pressure ratio versus Mach number

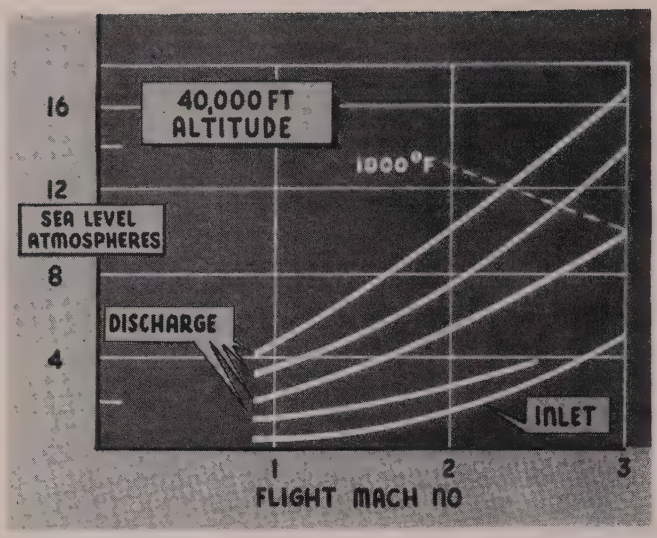

Fig. 8 – Absolute pressure versus Mach number

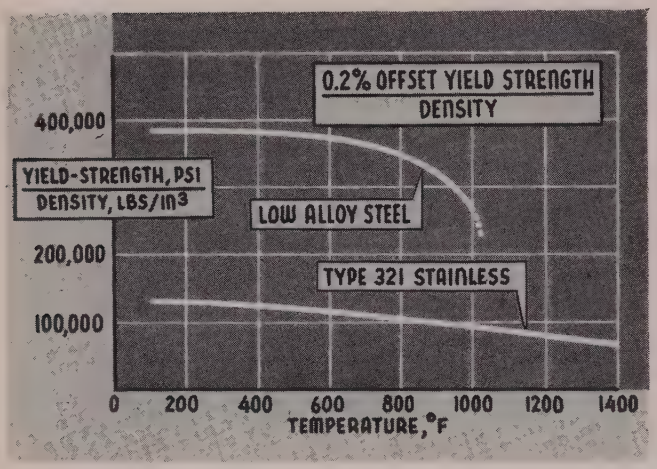

Fig. 9 – Yield strength/density ratio versus temperature

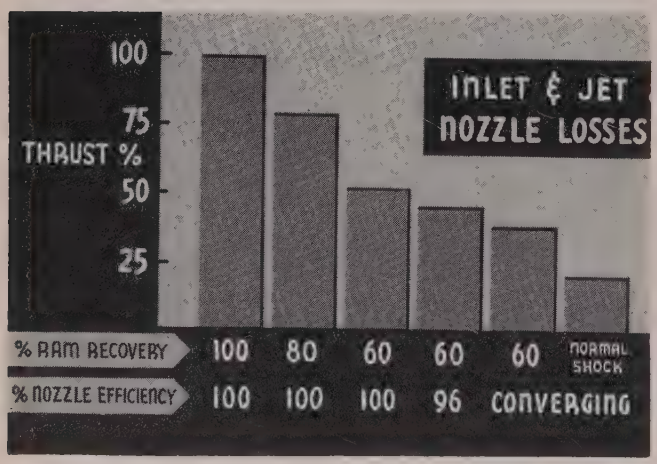

Fig. 10 – Inlet and jet nozzle losses at high flight Mach numbers

cumferential stress caused by the internal gas pressure, which is highest during high-speed flight.

The high compressor discharge temperatures will require all steel construction. Fig. 9 shows the ratio of yield strength to density plotted as the ordinate against temperature as the abscissa. The strength of the low-alloy steel is decreasing very rapidly at temperatures approaching 1000 F. The 321 stainless steel can be applied to higher temperatures, but it is seen that the ratio of yield strength to density is much lower than for the high-strength low-alloy steel. It is evident that substantial weight increases will result if the temperatures are so high as to require the extensive use of stainless steels in the compressor casings.

A problem that has become critical with high flight Mach number is that of variable-area inlets and variable-area exhaust nozzles. The problem is that of obtaining high efficiencies over a wide range of operation from take-off and climb to supersonic flight speeds at altitude. The penalties at high Mach numbers for poor ram recovery and exhaust nozzle efficiency are illustrated in Fig. 10.

The required inlet duct area variation for a typical engine is shown in Fig. 11. The throat area should be variable with Mach number, engine speed, and altitude. The required variations in exhaust nozzle throat and exit area for a typical engine are shown in Fig. 12. Both the throat exit areas show a required variation of about two to one. The losses that would be incurred with fixed area inlets are illustrated in Fig. 13, which shows the performance of two fixed area inlets, one designed for Mach 1 and the other for Mach 2. The fixed area inlet will not operate over a wide range of Mach numbers with good efficiency, and therefore the design must be made variable to match all conditions. A similar requirement exists for the exhaust nozzle, as shown in Fig. 14. The exhaust nozzle must be made convergent-divergent for the

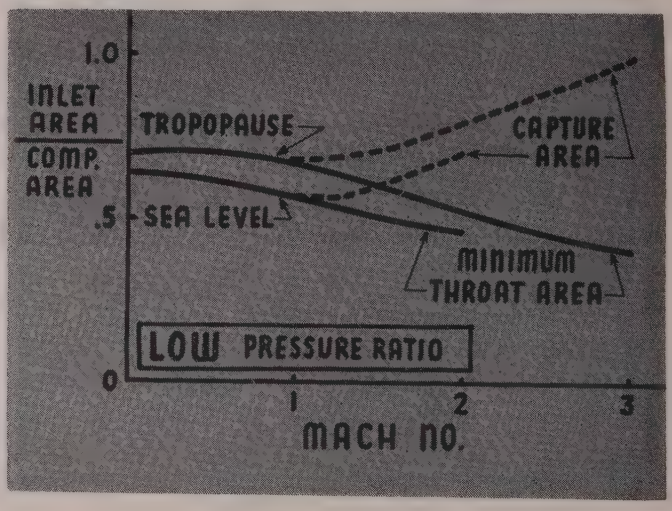

Fig. 11 – Required variation in inlet duct area for a typical engine

high expansion pressure ratios attained at high flight speed. Both the throat and exit areas must be made variable to accomplish the variations in Mach speeds and operation with and without reheat. A physical picture of the required change of jet nozzle configuration is illustrated in Fig. 15.

Summary and Conclusions

In summary, the jet engine designed to fly most of its missions between Mach 2 and Mach 3 has problems accentuated by the high temperature and pressure levels and the requirements for a wide range of operation aerodynamically. This will require careful attention to problems of stress and weight, bearing cooling, accessory cooling, and variable-area inlets and exhaust nozzles.

Some specific conclusions are:

1. Turbojet engines have good performance characteristics over a wide range of flight speeds extending well up into the supersonic speed range. Thermal efficiency improves at the higher speeds.

2. Inlet and exhaust nozzle losses have great effects on the performance of the installed powerplant. The jet nozzle problem is particularly critical because of the wide range of flow areas and pressure ratios.

3. For the highest flight speeds, insulation and cooling of the casings containing compressor discharge air may be necessary if materials having the highest ratios of strength to weight are to be used for the casings.

Acknowledgment

The authors gratefully acknowledge the valuable assistance given by W. H. Krase and Mason Campbell in the preparation of different parts of this paper.

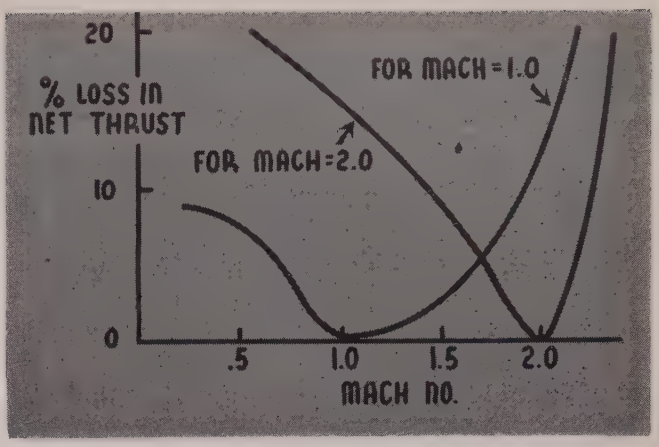

Fig. 13 – Thrust losses incurred with fixed area inlets

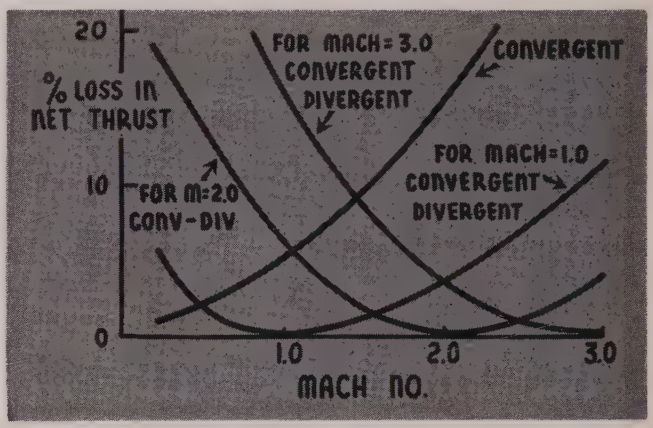

Fig. 14 – Thrust losses incurred with various exhaust nozzle designs

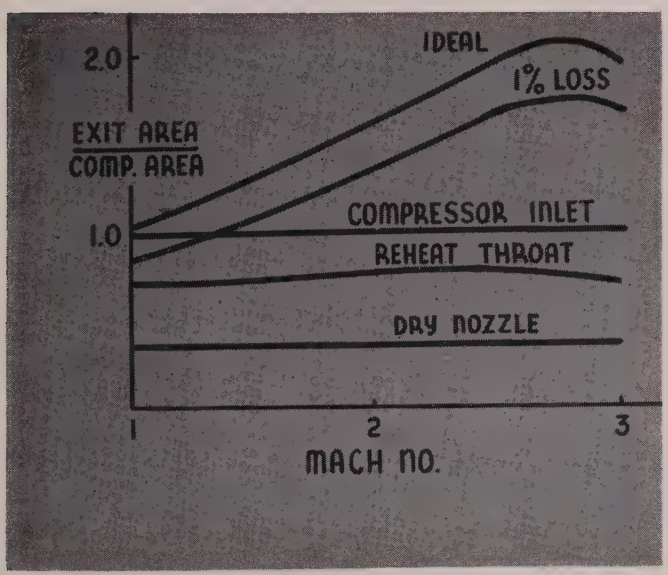

Fig. 12 – Required variation in exhaust nozzle throat and exit area for a typical engine

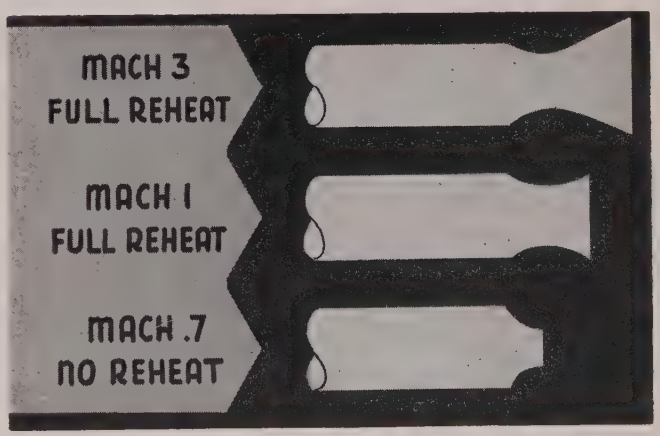

Fig. 15 – Required jet nozzle designs at various Mach numbers

Effects of Residual

EXPERIMENTAL analyses of residual stresses as related to fatigue properties and to the surface failure known as spalling are discussed in this paper.

How those processing operations such as heat-treatment, machining, grinding and straightening contribute the factors causing residual stresses and what are the practical methods of measuring these stresses are also described here.

The Author

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THE subject of this presentation is an old one, yet it is vitally new. It is an old one since it has interested scientists and engineers to the extent that a recent bibliography[1] lists about 1300 references, some dating back to 1911. It is vitally new because engineers today do not usually know much about the residual stresses existing in important highly loaded parts. They do not know how much these residual stresses vary from one part to the next. They have no established means of predicting the nature of the residual stress distribution in an actual part, and they have only limited means at their disposal for determining these stresses. To predict engineering properties it has been necessary to rely principally upon hardness determinations, microstructure, hardenability, chemical analyses, tensile tests, fatigue tests, and so forth. While each of these is important in its own right and each of these is related to residual stresses, none of these as presently interpreted reveals even a qualitative clue as to the residual stress state in any completed part. The reason for this lack of consideration of residual stresses over the years is undoubtedly the lack of a simple nondestructive means for measuring these stresses.

Before going further, however, it is perhaps fair to ask: "How important are these hidden stresses?" "Why worry about them?" Of course, the answer is that they are at least as important as stresses caused by externally applied loads. I would be inclined to think that they are even more important, because the distribution of stresses caused by external loads is relatively simple as compared with the triaxial internal force distribution caused by the combined mechanical and thermal history. Chemical composition, shape, size, heating, quenching, tempering, machining, grinding, polishing, lapping, etching, cleaning, plating, hydrogen absorption, stress-relieving – all leave their mark on the final residual stress pattern.

Trapped in many component parts are forces which only become manifest by parts which crack on the shelf or fail prematurely. Quench cracks, grinding cracks, stress corrosion cracks, all give testimony to these "unharnessed" forces.

In spite of all the work that has been done and the growing awareness of these hidden forces, the engineer is unfortunately in the position of not being able to determine quickly these stresses or to predict them satisfactorily on the basis of supplemental observations.

The various effects of residual stresses on the behavior of metals have been extensively documented. There is evidence of beneficial results by introducing residual compressive stresses in numerous parts. There is no point in reiterating these. They are well covered in the literature.

In this paper, I will discuss experimental analyses made of residual stresses as related to fatigue properties and as related to a surface failure in the form of spalling. Furthermore, the attempt is made to classify factors causing residual stresses and how processing operations fit into this classification. Finally, I will describe briefly the practical residual stress measurement techniques used in our laboratory. Included are a few interesting observations which contribute something to our knowledge of this subject.

Load Stresses and Residual Stresses

First, let's take a quick look at an oversimplification of the relationship between load stresses

[1] See International Harvester "Bibliography of Residual Stress," Originally compiled in 1951 by Armour Research Foundation for International Harvester Co.; revised and extended to June, 1953, by T. C. Huang of International Harvester Manufacturing Research Department.

Stress on Fatigue Life of Metals

R. L. Mattson, Research Laboratories Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1954

and residual stresses. Fig. 1 shows a simple beam stress diagram. The dashed line shows the typical stress distribution in simple bending. The shaded area shows an actual residual stress distribution found in a leaf-spring specimen. This stress existed in the piece of steel with no external forces applied. When this force distribution is added to the load-stress distribution, the result is as shown by the solid line. Note that the material suffers lower stress near the critical surface (top surface) because of the favorable residual compressive stress. Fatigue life was much greater because of this protective compressive stress at or near the surface.

Fatigue Life versus Residual Stresses

Although much has been published about the possible relationship between residual stresses and fatigue properties, actual experimental data on this relationship has been quite limited. From an accumulation of analyses of residual stresses in flat specimens of rectangular cross-section and unidirectional fatigue tests, we have obtained data on this important relationship. A detailed report of this investigation was presented to an SAE technical committee last October.[2] It is sufficient to say here that specimens similar to automotive leaf springs were fatigue tested in one-directional bending. Variations in residual stress were obtained by shot peening and by shot peening while the test surface was in a state of tension.

Important results of the fatigue tests and residual stress analyses are shown in Fig. 2. All specimens were cycled from approximately zero stress to 200,000-psi tensile stress. The specimens (nonpeened) which had the shortest fatigue life also had the lowest residual compressive stress near the surface. The intermediate group (shot peened) had longer life and also had a higher residual compressive stress. The group which had the longest life was shot peened while under strain (sometimes called "strain peening") and had the highest residual compressive stress. The relationship between residual stresses and fatigue properties is strikingly apparent.

Plotting this data and some supplementary data in a different way, we have the results shown by Fig. 3. Here we have plotted what we have chosen to call "net stress" versus fatigue life. Net stress is defined as the algebraic sum of load stress and the residual stress as shown by the upper right inset. The load stress at the surface for all of these specimens was held constant at 200,000 psi. Hence, the net stress was obtained by simply adding algebraically the peak residual compressive stress found in our analyses to the load stress at the corresponding depth. It may be seen that the nonpeened specimens which had little residual compressive stress have high "net stress" and a correspondingly low fatigue life. The shot-peened specimens had a medium value of residual compressive stress and hence a medium net stress and a correspondingly medium fatigue life. While those specimens peened while under strain had a high residual compressive stress, hence a low net stress and a high fatigue life. The net stress can be considered to be the maximum stress that the material is subjected to in a tensile direction, and hence it is related to the fatigue life. For all specimens, the stress range through which the material was subjected was substantially the same. Thus, taking into consideration both the stress caused by the

[2] See "Effect of Shotpeening Variables and Residuals on the Fatigue Life of Leaf Spring Specimens," by R. L. Mattson and W. S. Coleman, Jr., on pp. 546-556 of this issue.

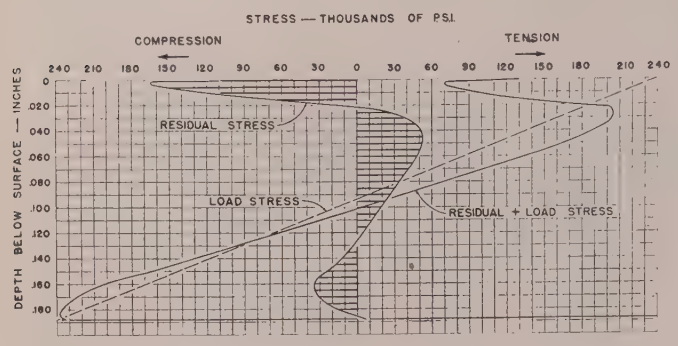

Fig. 1 – Combination of residual stress and bending stress. Stress through complete thickness of specimen is shown

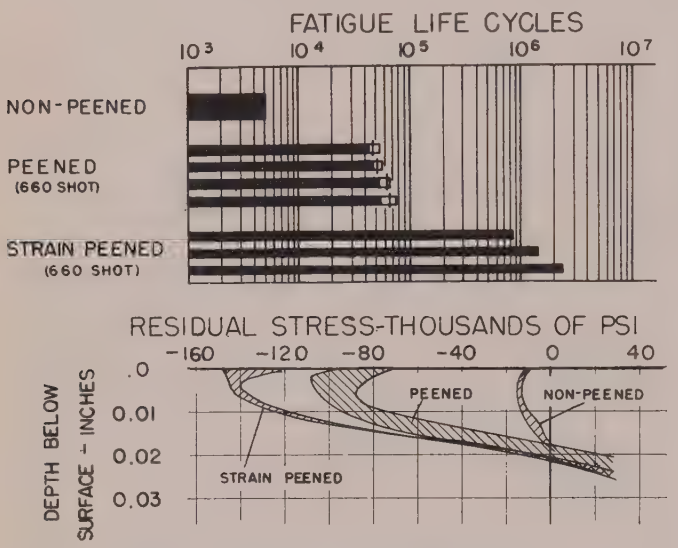

Fig. 2 – Correlation of fatigue life and residual stresses, one-direction bending, approximately 0 to 200,000 psi tension

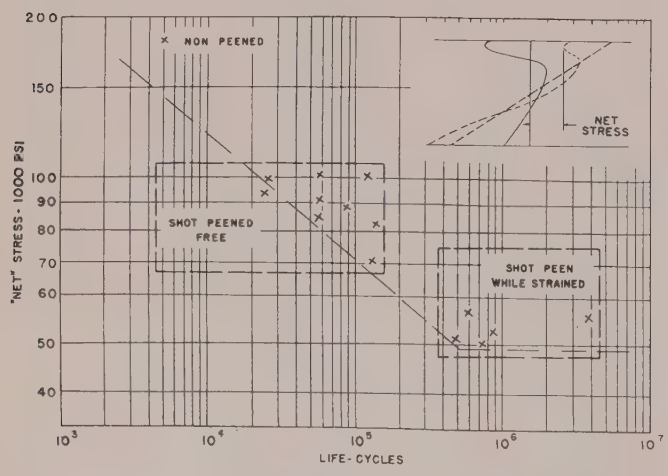

Fig. 3 – Plot of net stress versus life, where net stress equals load stress plus maximum residual stress

applied load and the internal or residual stress, we find something very similar to the typical SN curve which characterizes fatigue in metals. Therefore, we have experimental evidence that residual stresses do indeed affect fatigue properties and that residual stresses behave in a manner similar to stresses due to applied load. Although this relationship was anticipated by many, experimental data has been lacking.

In this investigation it was not presumed that work hardening did not occur and that work hardening was not a contributing factor. Work hardening is something rather difficult to measure, while residual stresses can be measured. Hence, experimental correlation is readily possible between fatigue properties and residual stresses, while such correlation is difficult between work hardening and fatigue life.

Spalling versus Residual Stress

An examination of residual stresses in relation to the spalling-type failure also proved to be enlightening. Spalling is considered here to be a surface fatigue phenomenon characterized by a flaking of a surface layer such as is sometimes encountered in gear contact surfaces. It might be considered as a special form of pitting. If the spalling-type failure occurs, the contention is that it is associated with a particular kind of residual stress distribution. As an example of spalling, a failed bearing ball is shown in Fig. 4. The fracture surface running substantially parallel to the original surface may be noted, and it is this characteristic which distinguishes spalling from pitting. An analysis of this bearing ball is shown in Fig. 5. This figure shows a cross-section of the failure alongside the tangential residual stress distribution existing in the ball. The failure apparently started in the region stressed in tension below the surface and propagated radially until it encountered a zone which was residually stressed in compression. Then the fracture veered off, running parallel to the surface, and subsequently broke through to the surface. It can be seen that the fracture path running parallel to the surface is located in the area of high stress gradient. In this zone, the residual stress ranges from a fairly high compressive stress to a tensile stress in a very short distance. The fracture finds it difficult to penetrate the layer of metal residually stressed in compression, but apparently the steep stress gradient plus the stresses imposed by contact with the race were sufficient to encourage propagation in a direction parallel to the surface.

In another example, the surface of an induction-hardened bearing was found to be full of cracks as shown in Fig. 6. These cracks extended vertically from the surface for a short distance and veered off, running parallel to the surface. Results of a residual stress analysis of an adjacent area are shown in Fig. 7. Here it can be seen that again

the spalling is associated with a severe stress gradient. There is a tensile residual stress at the surface and a compressive stress beneath it. Actually, since the surface had cracked, we would expect much higher residual tensile stresses than those measured. Here the fracture penetrated vertically until it encountered a layer of material residually stressed in compression and then traveled parallel to the surface. It is to be noted again that the fracture running parallel to the surface was also in a zone of a steep stress gradient. In this case the surface tensile stress was attributed to the grinding operation.

From these examples, we come to the tentative conclusion that if we observe a surface failure characterized by fracture planes parallel to the surface, we might expect a high residual stress gradient in that area as shown in Fig. 7. If we observe vertical fracture planes in a surface layer in combination with a fracture plane parallel to the surface, we would be suspicious of a steep stress gradient with the surface layer in tension and the material below in compression as shown by the upper part of Fig. 8. With subsurface fracture planes parallel to the surface and vertical planes below, we would expect a surface layer stressed in compression and tensile stresses below as shown by the lower part of Fig. 8.

Processing Operations versus Residual Stresses

In General – Since residual stresses are important to the mechanical properties of our parts, it is important to know something about how they come to exist. Any process which causes localized plastic flow or localized phase transformation of the material will set up or alter its residual stresses. This of course means residual stresses develop or are altered by various steps in heat-treatment, by machining, grinding, cold working, straightening, static loading, and fatigue loading. With such an array of influencing factors, one can

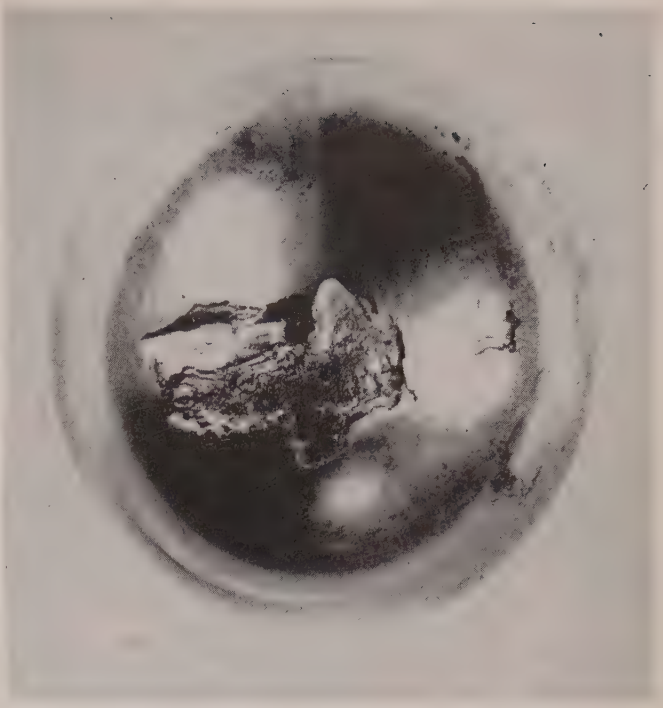

Fig. 4 – Spalling-type failure in 13/16-in. diameter bearing ball

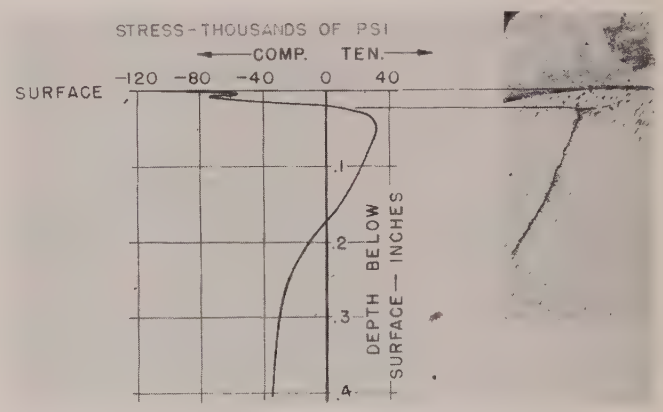

Fig. 5 – Typical residual stress pattern of used 13/16-in. diameter ball

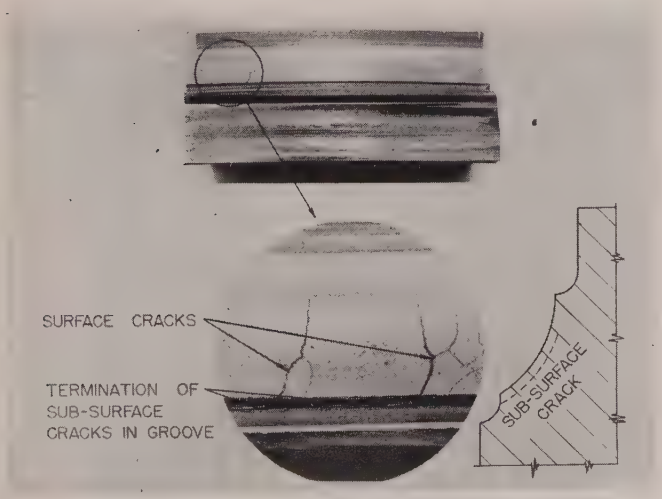

Fig. 6 – Example of spalling failure in large bearing

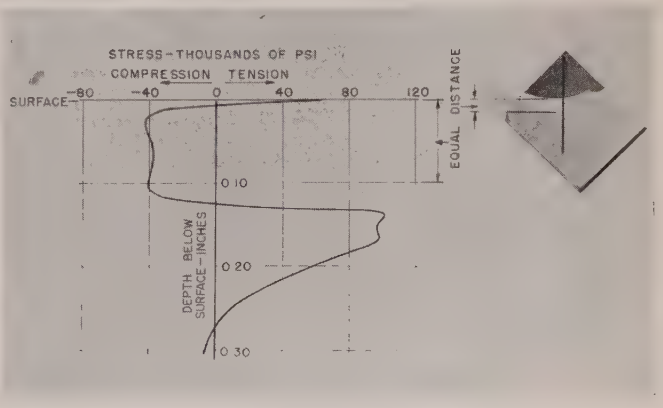

Fig. 7 – Residual stress in large bearing race

Table 1 – Types of Residual Stress Introduced by Processing Operations

Residual Stress Introduced or Altered By	Temperature Difference	Phase Transformation	Cold Plastic Flow	Warpage	Altered Chemistry
Heat-treatment (quenching and tempering)	Yes	Yes	No	No	No
Case hardening (carburizing and nitriding)	Yes	Yes	No	No	Yes
Induction hardening	Yes	Yes	No	No	No
Machining	Possibly	Unknown	Yes	Yes	No
Grinding	Yes	Unknown	Yes	Yes	No
Cold working (shot peening, surface rolling, tumbling, honing, lapping, blast cleaning)	No	No	Yes	Yes	No
Straightening	No	No	Yes	No	No

see the complexity of the problem of predicting residual stresses in fabricated parts. Some of these operations are listed for your examination in Table 1 together with a classification of the mechanism responsible for the residual stresses.

Heat-Treatment Stresses – The effects of chemical composition, size, shape, and heat-treatment alone constitute a tremendous problem and will not be dealt with in this paper, but there are many references in the literature on this subject. It should be mentioned, however, to emphasize the magnitude of the problem, that four steels within one hardenability specification, or H band, gave different residual stresses.[3] Hence, although there may well be a significant relationship between hardenability and residual stresses, many other factors must be considered. In spite of the size of this problem, I believe the solution is far from hopeless. Residual stresses introduced (or altered) by heat-treatment are generally divided into two classes which can at times work in opposing directions. These are called thermal stresses and transformation stresses. Thermal stresses are caused by temperature differences between, say, the inside and the outside of a component. Residual stresses result if at any point the stresses caused by temperature difference exceeds the yield point at that temperature, hence causing localized plastic flow. If thermal stresses alone were operating, the quenching operation would introduce in the surface of a part a state of residual compression. The transformation stresses are caused by phase transformation and are brought about because changes from one phase to another often entail a volume change. Martensite occupies more space than the austenite from which it comes. Hence, it requires more room, imposing stresses on neighboring areas as well as within itself. Another complication arises in that the transformation itself is undoubtedly affected by the transitional stress state. The combination of thermal stresses and transformation stresses produces the final stress state which can often be tensile at a critical surface, but it could just as well be compressive if we knew how to produce it.

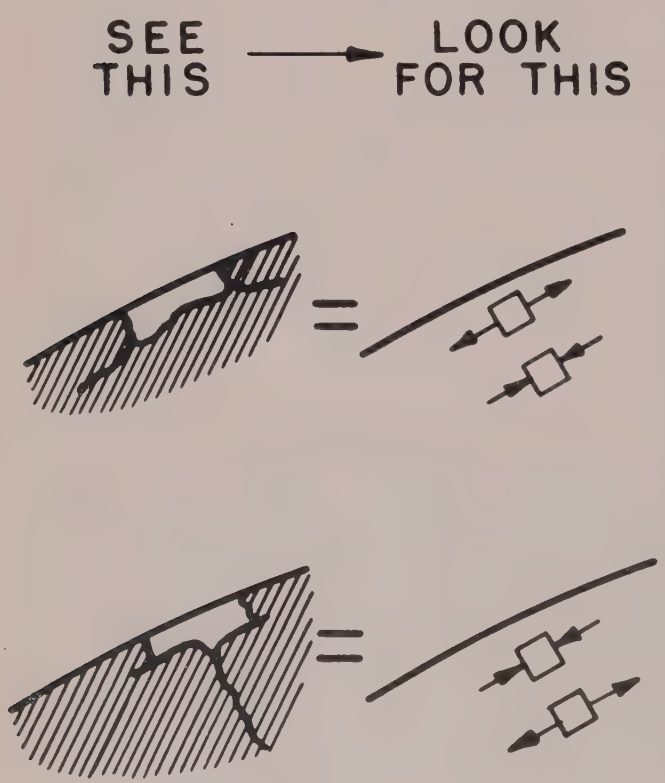

Fig. 8 – Spalling versus residual stress

Machining Stresses – Very little experimental data is available on the nature of the residual stresses introduced by machining operations. The variety of metal removal methods suggests that considerable variation is possible in the stresses trapped in the surface layers of the part. In Table 1 the opinion is expressed that several factors are involved in the introduction of residual stresses during machining. Trapped stresses can be introduced by high mechanical stresses associated with the severing of the metal. These in turn are associated with the geometry of the cutting tool, speed of cutting, nature of cutting, and so forth. If sufficiently high temperatures are developed locally, thermal stresses or even transformation stresses might be introduced. Also in machining, if the metal removed was originally stressed, some unexpected change in shape or size would occur, causing a change in the residual stress in the remainder.

Grinding Stresses – Grinding is one of our most important fabricating processes, and it is also one of our important residual stress makers. Everyone knows about "grinding cracks," which of course are simply manifestations of high tensile stresses

[3] See *Metal Progress*, Vol. 57, February, 1950, pp. 183-188: "Some Effects of Quenching and Tempering on Residual Stresses in Steel," by A. L. Boegehold.

introduced by grinding. It is easy to see how grinding introduces residual stresses. It would be much more difficult to explain how grinding might not introduce such stresses. The combination of stresses imposed by the grain of the wheel rubbing and cutting the metal, the heat generated locally producing high temperature gradients, localized plastic flow, and the metallurgical phase changes which might take place causes the resultant residual stress. Surface temperatures as high as 3000 F due to grinding have been calculated and reported.[4] Several years ago our laboratories made an analysis of residual stresses caused by grinding which showed enormously high residual tensile stresses at the surface.[5] More recent work in this field by others reveals some important findings. The experimental data on residual stresses due to grinding, shown in Fig. 9, is from the work of H. R. Letner and H. J. Snyder of the Mellon Institute. Biaxial stresses were measured in fully annealed manganese oil-hardening tool steel for various depths of cut. Results of the measurements are shown for one depth of cut for both longitudinal and transverse directions of abrasive travel. In a longitudinal direction there is a layer of material stressed in tension, but in a transverse direction, this surface layer is stressed in compression. So it cannot be categorically said that grinding introduces only tensile stresses. The astonishing thing about this data is the steep stress gradient. Here a tensile stress exists at the surface which is almost as high as the ultimate strength of the material, and within 0.0002 in. the residual stress is compressive! The observation that at one point the stress can be near the ultimate strength in tension and at point 1/5 of 0.001 in. away the stress is different by 100,000 psi is truly amazing. Fig. 10 shows a plot of the residual stress at the surface only versus the depth of cut for both stresses in the direction of the grinding marks and perpendicular to the grinding marks. It may be seen that the stresses parallel to the grinding marks approach the ultimate tensile strength shown by the dashed line as the depth of cut increases. It also can be seen that it does not take a very deep cut to introduce large tensile stresses. Furthermore, the stresses perpendicular to the grinding marks, at least in this case, are compressive at the surface for all depths of cut shown.

Glikman, Sanfirova, and Stepanov[6] found residual stresses due to grinding in low-carbon steel and high-chromium stainless steel as shown in Fig. 11. In low-carbon steel, when grinding with the side

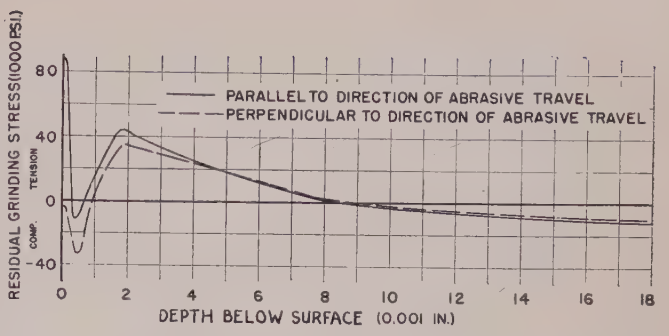

Fig. 9 – Residual stress distribution in annealed manganese oil-hardening tool steel due to surface grinding (from Letner and Snyder)

of the wheel, residual tensile stresses ranging from approximately 35,000 to 50,000 psi were found at the surface. These stresses extend to depths between 0.004 to 0.012 in. Only one curve for low-carbon steel is shown in Fig. 11 for simplicity. Results similar to Letner and Snyder's were found in high-chromium stainless steel with an intermediate layer in residual compression in some cases. Of particular importance is a comparison between two grinding techniques, one with the OD of the wheel and one with the side of the wheel on high-chromium stainless steel. Grinding with the OD of the wheel produced a surface layer 0.001 in. deep stressed in tension, while grinding with the side of the wheel produced a surface layer 0.015 in. deep stressed in tension. This difference suggests that the larger the contact area between the grinding wheel and the work, the greater will be the volume of material stressed in tension. The extent of damage is apparently less for small contact areas even though actual tensile stresses at the surface are about the same.

Similar data on hardened steels is not available but would be most useful since most critically

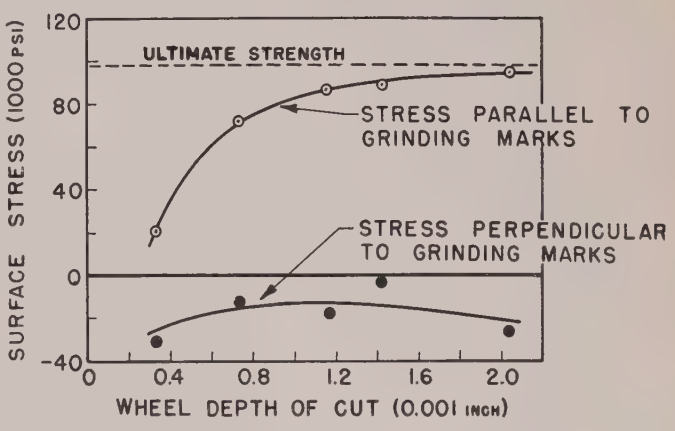

Fig. 10 – Residual stress at the surface introduced by grinding (from Letner and Snyder)

[4] See ASME Transactions, Vol. 74, January, 1952, pp. 73-86: "Surface Temperatures in Grinding," by J. O. Outwater and M. C. Shaw.

[5] See *Metal Progress*, Vol. 43, February, 1943, pp. 209-15, 270: "Peened Surfaces Improve Endurance of Machine Parts," by J. O. Almen.

[6] See *Journal of Technical Physics* (USSR), Vol. 19, No. 4, 1949, pp. 441-447: "Residual Stresses Set Up in High-Chromium Stainless Steel by Grinding," by L. A. Glikman, T. P. Sanfirova, and V. A. Stepanov. (Brutcher Technical Translation No. 2562.)

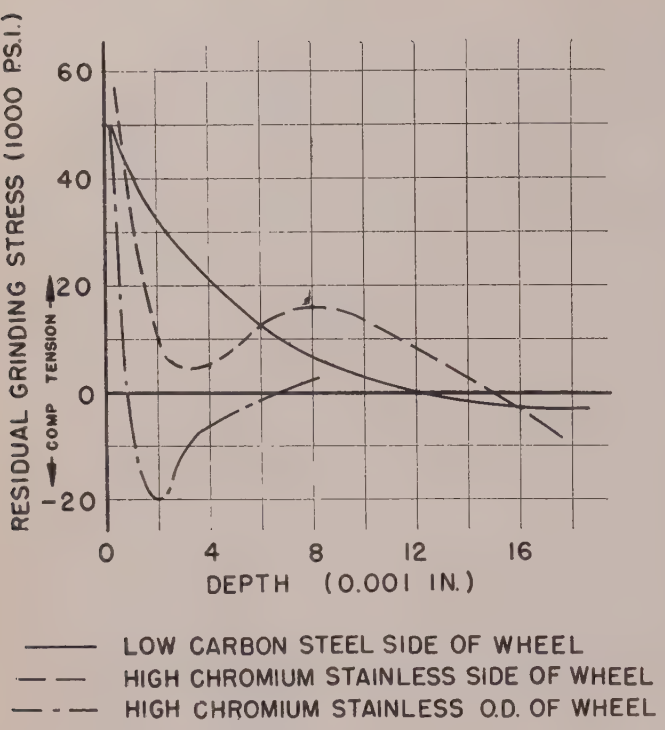

Fig. 11 – Residual stresses caused by ordinary shop grinding (from Glikman, Sanfirova, and Stepanov)

loaded parts are much harder than those investigated. So we can only use conjecture in applying this information to practical problems. It seems in order, however, that it should be reemphasized that we should be very cautious about grinding parts in critical areas because of the potentiality of introducing damaging residual tensile stresses. Light cuts are to be preferred to heavy cuts, and minimum wheel contact area with the work is to be preferred over broad contact area.

(e) *Cold Working versus Residual Stresses* – In cold-working processes such as shot peening and surface rolling, residual compressive stresses are introduced in a surface layer. Some indication of the range of residual compressive stress introduced by surface peening is shown in Fig. 12. Here compressive stresses as high as 180,000 psi and 0.010 in. deep were measured in a shot-peened ball-bearing race, which of course is very hard, being over Rockwell C 60. Compressive stresses in the order of 100,000 psi and 0.025 in. deep were introduced by shot peening leaf-spring specimens of hardness Rockwell C 46. In annealed SAE 1020 steel, residual compressive stresses of 40,000 psi and 0.35 in. deep were introduced by air hammering with a spherical-nosed tool. In these cold-working processes, there is apparently a relationship between the maximum residual compressive stress introduced and the yield strength of the material. This would likely be influenced by the work-hardening tendency of the material.

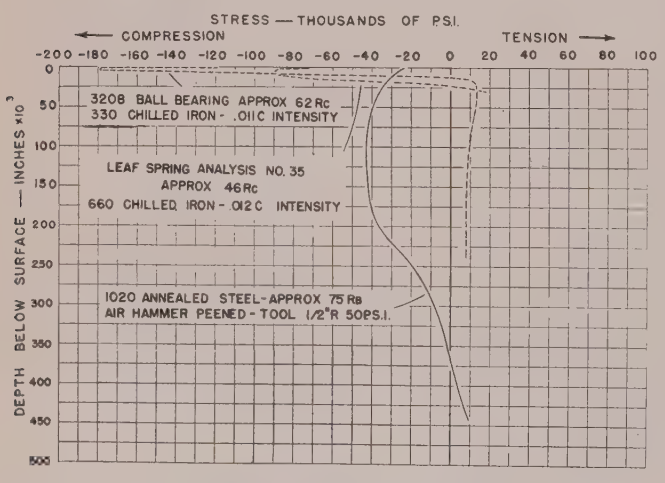

Fig. 12 – Effect of hardness and intensity of cold working on magnitude and depth of compressive layer

Some experiments were conducted to determine the residual stresses introduced by tumbling with alundum abrasive, which is certainly a very mild form of cold working. The results are shown in Fig. 13. Here it can be seen that residual compressive stresses in the order of 70,000 psi are introduced in a surface layer and that these extend to a depth of 0.002 in. Tumbling time also affects the amount of compressive stress introduced.

Contrasting this with grinding, where tensile stresses are introduced, suggests that some form of cold-working process is desirable as a final operation for parts subjected to fatigue loading. Comparative data has already been reported by Tarasov and Grover[7] indicating the superiority of such a cold-working process over grinding on fatigue properties.

Straightening versus Residual Stress – One operation with which all of you are familiar is the straightening operation. Since in the operation of straightening certain portions of a cross-section suffers plastic flow while other portions are strained elastically, residual stresses are introduced. By bending beyond the yield point, as can be seen in Fig. 14, material near the surface yields, and upon removal of the load, the bar returns to an equilibrium position with the surface that was loaded in tension now residually stressed in compression and the surface that was loaded in compression now residually stressed in tension. From fatigue tests on certain straightened axle shafts, it has been observed that the failure always occurred on the side which was residually stressed in tension by bend straightening.[8] Straightening by selective surface hammering or by selective surface rolling, in which residual compressive stresses are introduced, pro-

[7] See ASTM Proceedings, Vol. 50, 1950, pp. 668-698: "Effects of Grinding and Other Finishing Processes on the Fatigue Strength of Hardened Steel," by L. P. Tarasov and H. J. Grover.

[8] Private communication from C. F. Double, Chevrolet Motor Division, GMC.

vides shafts of improved fatigue durability. This type of straightening introduces favorable compressive stresses in both surfaces.

Measurement of Residual Stresses

Since these internal forces, or residual stresses, are important, the measurement of their magnitude is extremely desirable and, in some instances, absolutely necessary. Although we cannot accurately predict them at the present time, we can at least make qualitative determinations of them, and, if we want to, we can even make fairly accurate quantitative measurements of them. However, for the practical engineer or metallurgist, the problem of measuring residual stresses often seems out of the question. Yet, he is sometimes called upon to explain failures that cannot be explained after the usual metallographic examinations. He may suspect the existence of unfavorable residual stresses, but he can't be sure unless he makes a residual stress analysis.

There are a number of techniques available to him for making residual stress analyses. Most of them have been described in the literature. Each has its limitations and advantages. The X-ray diffraction method, for example, is the only nondestructive method so far described. This technique has now been developed to the point where residual stresses may be measured in materials with Rockwell "C" scale readings up to 60. The main drawbacks of this method are costly equipment and the requirement of a smooth, flat surface.

Among the qualitative techniques is that of drilling a hole in the specimen and then noting the relaxation of the stresses in the adjacent metal by means of brittle lacquer or strain gages. But the stresses introduced by the drilling may sometimes make the results obtained questionable.

Another qualitative technique commonly used is the "split-deflection technique." This technique is one in which the part to be studied is cut through some section and then the direction and amount of deflection measured. A ring, for example, may be split radially and after splitting may spring outward making a larger diameter, as shown in Fig. 15, or spring inward making a smaller diameter. From the observation of these deflections, the conclusion is often drawn that one surface was in tension and the other in compression. This is not true unless there is a simple residual stress distribution, but we often find residual stress distributions that are "U" or "S" shaped or that have one or more inflection points. If the residual stress distribution is complicated, then any conclusions about the surface stresses drawn from such deflection observations are meaningless. So, the split-deflection technique should only be used when the assumption can safely be made that the residual stress pattern is simple.

Dissection Techniques

The most accurate residual stress measurement techniques are the "dissection techniques." In principle, the part or specimen is analyzed by taking strain measurements as known layers of materials are removed from selected areas by careful grinding, etching, or electropolishing. These strain measurements, of course, indicate the deformation undergone by the specimen due to the removal of the layer. The forces required to overcome this deformation are then calculated. These forces are equal to the forces that were present in the layer of material removed. Therefore, having determined

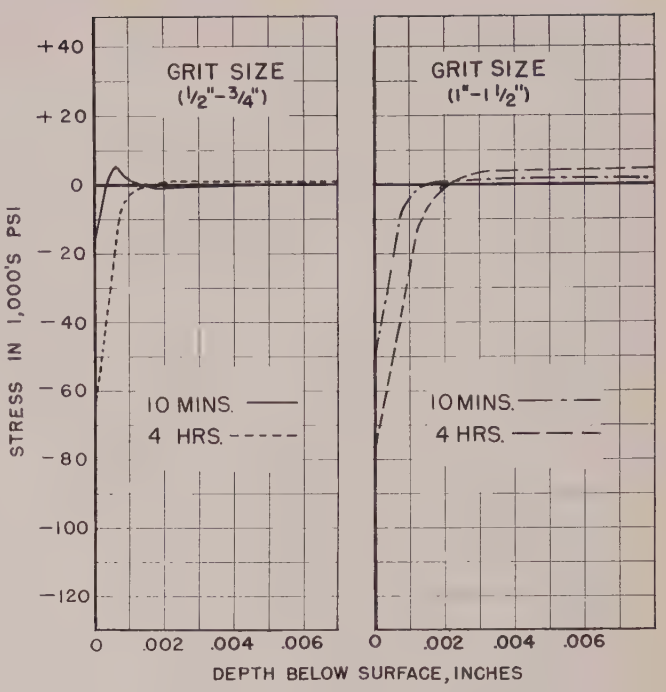

Fig. 13 – Residual stress introduced by alundum abrasive tumbling for various times with two grit sizes

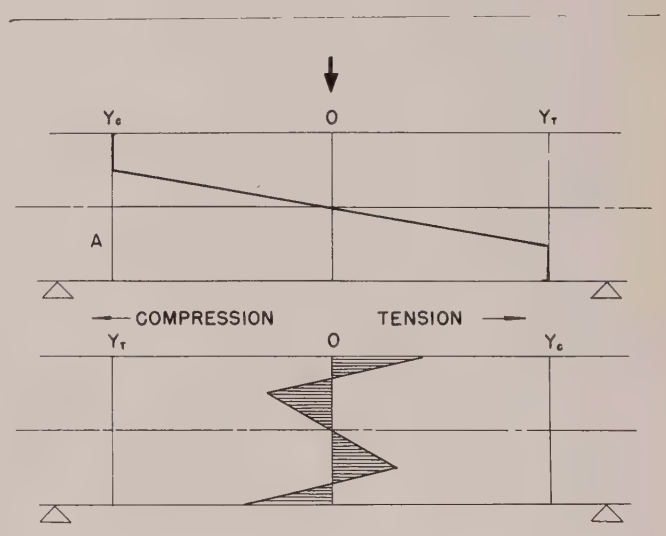

Fig. 14 – Induced residual stress in beam cold straightened in one direction only

these forces, the stresses in the layer removed may be calculated.

The Sachs boring-out method is a good example of the dissection technique. In this method, specimens of cylindrical shape are dissected by boring out layers from the inside. Measurements are made of the change in length and the change in diameter as each layer is removed. A variation of this method is to bore a hole in the specimen and then to remove layers from the outside surface, either by turning on a lathe or grinding, and to measure the changes in the dimensions of the hole. The Sachs method and the variation are basic and give the average longitudinal, tangential, and radial stresses in the layers removed.

In practical applications, we are generally confronted with the problem of determining the residual stresses in a component part, and we are usually concerned with a particular location on this part. Most of the time we are not interested in the stresses at great depths below the surface unless we know that the failure originates at some subsurface location. Hence, efforts have been directed toward developing techniques of residual stress analysis with this in mind.

At our laboratory we use the dissection method, but to simplify the analysis, we measure the stresses in only one direction at a time. This gives us satisfactory stress distribution measurement in the direction of principal interest. Essentially, we try to reduce each problem to either a simple beam of rectangular cross-section, in which we can easily measure the longitudinal stress distribution, or a ring of rectangular cross-section, in which we can easily measure the tangential stress distribution.

When this method is applied to a machine element, it is necessary to determine the stresses that are relieved when the specimen is removed from its parent part. This is done by noting the deformations that take place in the specimen as it is removed, and then with this information, calculating the stresses relieved. In the case of the beam-type specimen, the deformations will be a change in length and a change in curvature. In the case of a ring-type specimen, the deformation will be a change in diameter.

Fig. 16 shows a beam-type specimen and its parent part, a gear tooth. Also shown in the figure are the strain gages used to measure the change in surface strain and the arc-height indicator used to measure the curvature change when the specimen is removed from its parent part. These two measurements make it possible to determine the stresses that are relieved. This example also demonstrates how a rather complex part can be reduced to a simple geometrical shape that is amenable to calculations.

After the specimen has been very carefully removed from the parent part so that the minimum amount of stresses are introduced by this operation, it is ready for dissection. Layers of uniform thickness are removed by extremely careful grinding, which we have come to call "stress grinding." An outline of stress grinding is shown in Table 2.

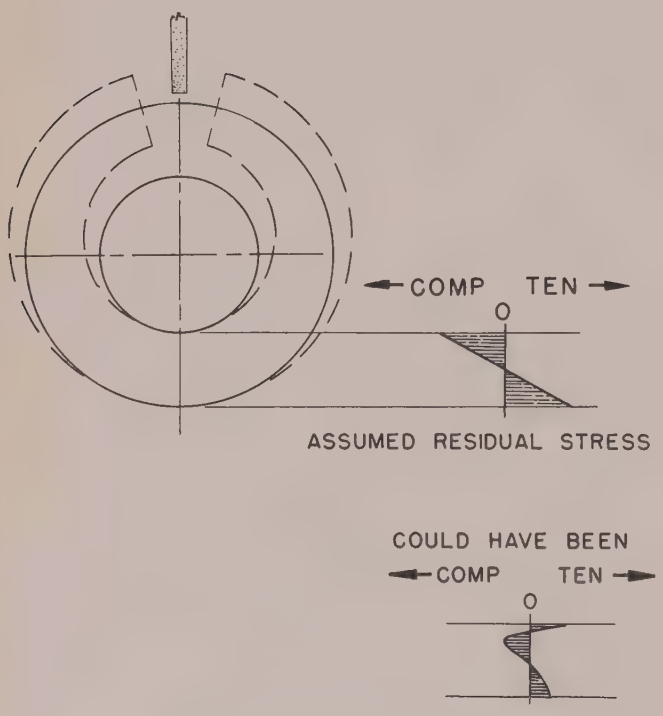

Fig. 15 – Split-deflection technique

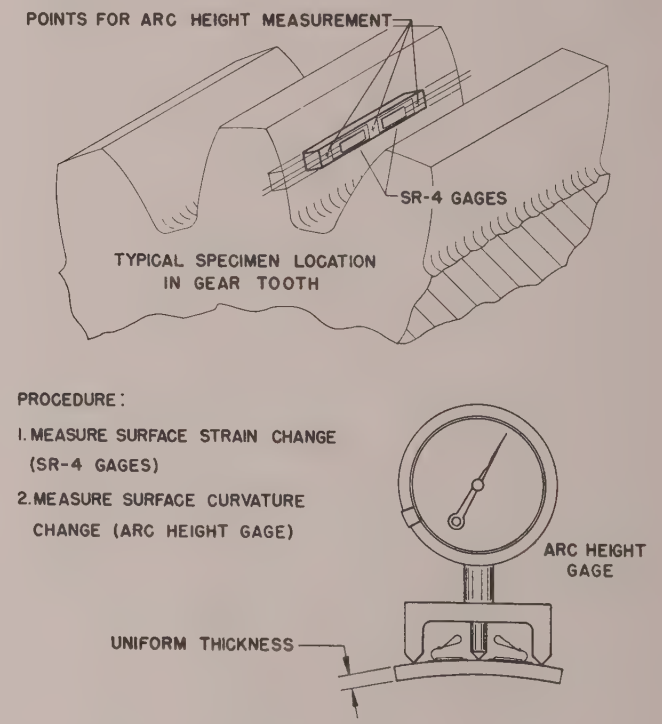

Fig. 16 – Determination of stress change from parent part to specimen

After each layer is removed, the change in curvature is determined by measuring the arc height in the case of the beam-type specimens, or the change in diameter is measured in the case of a ring-type specimen. The equations we use for the solution of the beam-type residual stress problems do not require that we know the change in length as each layer is removed, so only the curvature-change measurement is made.

In making an analysis of a simple beam specimen, the question arose as to which method of measurement of the change in curvature would give the better accuracy, strain-gage measurements or arc-height measurements. In answer to this question, a set of curves were prepared illustrating which of the two is the more accurate for a given specimen size. A summary of these curves is shown in Fig. 17. Sensitivities were taken as follows: ±50 microin. for the arc-height measurements and ±10 microin. per in. for the strain-gage measurements. These curves can also be used to decide the thickness of the layer to be removed between curvature determinations for a given acceptable error in stress for a given specimen size. The ordinate of these curves is based on a modulus of elasticity of 30 million. The curves can be used for materials of any modulus, however, by simply multiplying the ordinate by the ratio of the new modulus to 30 million.

The formulas used for making the residual stress calculations are available in the literature.[9, 10] Presently, our calculations are done by an IBM digital computer. Our only desk job is to plot a smooth curve through the data on thickness of metal removed and arc height or diameter at that thickness, and to tabulate these data for equal increments of metal removed. From the IBM computer, we obtain a tabulation of residual stress versus the depth below the surface of the specimen.

Table 2 – Typical Grinding Procedure for Removing 0.010-In. Layer of Metal

1. Dress wheel.
2. Grind first 0.005 in. in steps of 0.0003 in. deep.
3. Grind next 0.003 in. in steps of 0.0002 in. deep.
4. Grind last 0.002 in. in steps of 0.0001 in. deep.

Wheel Type: Medium soft, vitrified bonded No. 46 size grain, and of medium structure. (Example: Norton Co. No. 32 A 46 H 8 VBE.)

Wheel Peripheral Speed: 6400 sfm.

Redress wheel to keep sharp. (Example: After five passes of steps 0.0001 in. deep on specimen ¾ x 3 in. redressing is necessary.)

Cross Feed: 0.015 to 0.020 in. per pass.

More Work Needed in the Field

With so much unknown about residual stresses and so much benefit to be gained by additional knowledge in this field, it is undoubtedly one of the most potent fields of research in engineering materials today. The work, however, need not be confined to a few large laboratories. Many engineers and metallurgists are capable and qualified, but I rather suspect they may be hesitant. Once it is appreciated that these analyses are not as complex as might be believed, more practical searching for critically important residual stresses will begin.

[9] See Proceedings of Society for Experimental Stress Analysis, Vol. III, No. 1, 1945, pp. 40-61: "A Study of Certain Mechanically Induced Residual Stresses," by D. G. Richards.

[10] See Proceedings of Society for Experimental Stress Analysis, Vol. IX, No. 2, 1952, pp. 133-140: "Techniques in Residual Stress Analysis," by Walter Leaf.

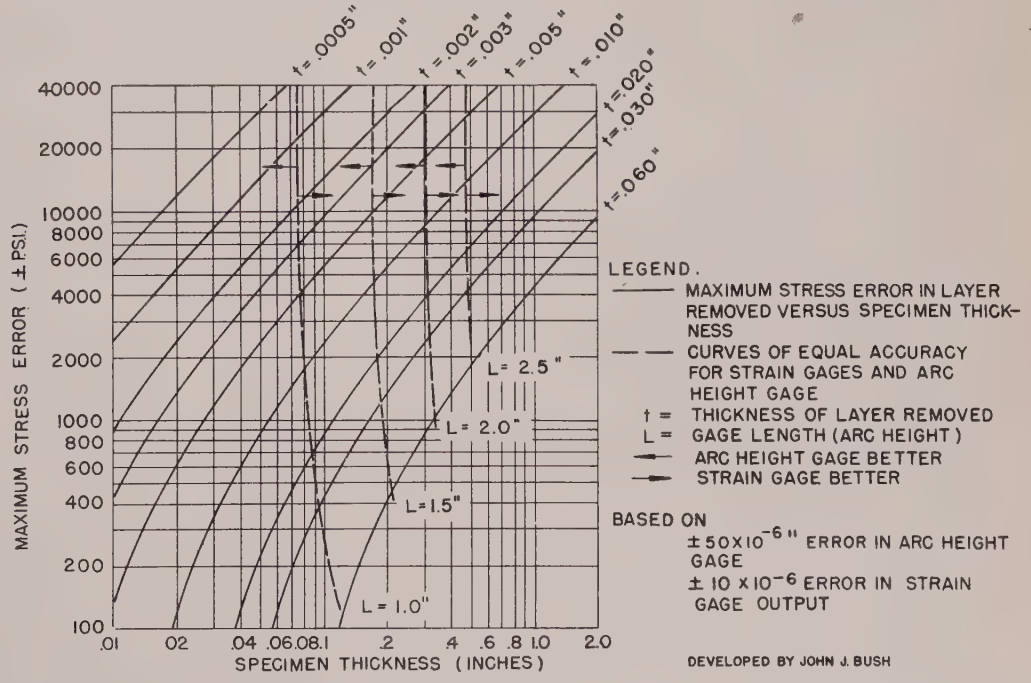

Fig. 17 – Error in residual stress analyses of simple beam specimen, specimen thickness versus layer thickness, arc height versus SR-4 strain gages

Free-Piston and Turbine Compound

PRESENT-DAY reciprocating internal-combustion engines – both the autoignition and the spark-ignition types – are the result of over 50 years of intensive evolutionary progress. Nevertheless, in the aircraft application the gas turbine powerplant is making a successful bid to displace the conventional reciprocating engine. This is so, even though the specific fuel consumption is of the order of twice that of the reciprocating-engine-propeller system! The fuel economy disadvantage is more than offset, at least for most military applications, by gains in power compactness, aerodynamic shape, specific weight, and basic mechanical simplicity. In other areas the gas turbine is also appearing as a strong competitor with the reciprocating engine, but not to such a spectacular extent as in aircraft.

How may the reciprocating internal-combustion engine acquire a better competitive position? One answer is that a "predestined marriage" should be accomplished between the reciprocating and the turbine type of engine. One major disadvantage of the gas turbine is the continuous-flow combustion process at a low temperature, achieved by high dilution (high excess air). In contrast, in the reciprocating engine there is a periodic combustion process at high temperature achieved with low dilution. The reciprocating engine, however, suffers from a poor utilization of the blowdown energy available after the power stroke is concluded, and the disadvantage of a relatively low-speed driveshaft and a poor torque-speed characteristic compared to a turbine. Thus, this marriage can result in substantial gains, provided the mechanical complexity is not excessive.

This union has already been consummated to a considerable degree, as evidenced by the commercial success of turbosupercharged engines, and more recently by the advent of the Wright Turbo-Compound (18 cylinders, 3 turbines) 30-W model. This engine, by deriving a substantial amount of power from the turbine components (15-20%), has a 10% lower specific weight and a 20% lower cruise specific fuel consumption. These are the very marked gains realized from a *revolutionary* concept in engine design, as compared to the inching progress obtained in the last 15 years by the normal *evolutionary* development of the reciprocating aircraft engine.

The purpose of this paper is to consider a more extreme concept of compound-engine design as incorporated in the free-piston gas-generator turbine combination. In this system the reciprocating-engine work is all absorbed by the supercharger requirement, and the high back-pressure flow of exhaust gas is the motive fluid for a power gas turbine. Under these circumstances there is no need mechanically to connect the turbine and

THE history of the free-piston and turbine compound engine development is reviewed briefly. After consideration of the status here and abroad, the salient features of the free-piston engine concept are considered. These are mechanical simplicity, compactness, and an excellent torque-speed characteristic, coupled with a fuel economy comparable to the conventional diesel, the modern record holder in this respect.

Some prognostications for the future of the development in this country are also presented.

The Author

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Engine—Status of the Development

A. L. London, Stanford University

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1954.

reciprocating engine components, and consequently a most advantageous fluid coupling is realized. Moreover, if a reciprocating-piston-cylinder type of supercharger compressor is employed, the engine crank can be dispensed with, and the compressor can directly absorb the reciprocating work of the internal-combustion-engine pistons. This simplification is accomplished by building the compressor and engine piston into a single assembly. Such an arrangement is shown schematically in Fig. 1a.

While the paper is concerned primarily with this compound internal-combustion-engine and turbine prime-mover version of the free-piston concept, the diesel-compressor version has played a very important role in the development and will, therefore, be considered briefly. This system, for the production of compressed air for pneumatic purposes, is shown schematically by Fig. 1b. Here the diesel-engine section has only a nominal supercharge (2-5 psig), sufficient for scavenging the 2-stroke cycle, and the compressor-airflow path is completely separate from the engine-airflow path.

The specific objectives of this paper are as follows: (1) To trace briefly the history of the free-piston-engine development in Europe, Asia, and in this country; (2) to summarize the important characteristics of the compound-engine system and to provide literature references for more detailed considerations of this nature; and (3) to make some prognostications regarding the future of the free-piston and turbine compound engine.

History of the Development

France – Marquis R. P. Pescara conceived the basic free-piston concept in 1922 while searching for a light-weight engine-compressor system to provide compressed air for helicopter rotor tip propulsion units.[1] The fact that only recently a great deal of effort has been devoted to a similar type of helicopter is a credit to the farsightedness of this inventor. Apparently his search for a commercially available light-weight engine-compressor combination was fruitless, so that, with the typical optimism of an inventor, he set about developing his own design. The result was a free-piston diesel-compressor system similar to that described in Fig. 1b. Shortly thereafter the idea was conceived of using the compressor to supercharge the diesel and to exhaust the diesel at a high back pressure into a power gas turbine such as indicated in Fig. 1a.

This concept became the focal point of Pescara's efforts, and he assembled a capable group of designers and engineers whose intensive efforts over the past 28 years have brought to commercial fruition both the compressor and prime-mover versions of the free-piston engine. Chief among these collaborators is R. Huber, a student of Stodola's, who joined Pescara shortly after graduation.[2] Mr. Huber is now the technical director of the Société d'Etudes Méchaniques et Énergétiques, which is the design organization for the Société

[1] See *Journal de la SIA*, Vol. 10, 1937, pp. 423-433: "Free-Piston Machines" (in French), by R. P. Pescara.

[2] See *Schweizerische Bauzeitung*, Vol. 66, Nov. 27, 1948, pp. 661-667; Dec. 4, 1948, pp. 673-679: "Free-Piston Generators" (in German), by G. Eichelberg.

Industrielle Général de Mechanique Appliquée (SIGMA), the manufacturing organization.

The following models are in commercial production by SIGMA: model GS-34, 1000 shp (equivalent) gas generator and model P-13, 45 hp (equivalent) diesel-compressor set.

The very considerable sale of these 85-psig, 200-cfm, dolly-mounted, portable compressors helped to finance the costly and time-consuming effort required for the development of the highly supercharged gas-generator system. One may venture the estimate that if the compressor had not met with commercial acceptance the prime-mover development would not have succeeded.

It should also be mentioned that G. Eichelberg of Zurich, who served as a consultant for SIGMA and Pescara, contributed greatly to the technical success of the French program, as did D. Coste, a marine engineer, who recognized the potential of the free-piston and turbine compound engine as a marine and naval prime mover. Financial support from the French Admiralty was also instrumental in achieving final success.

In addition to about 1200 compressors in commercial use, the following model GS-34 gas-generator-turbine installations have been made:

(a) Central station applications[3]: A 2-generator, single-turbine 1500-kw installation at the SIGMA plant at Venissieux. A 2-generator, single-turbine 1250-kw installation for the Centrale Electric of France at Reims. Two other similar installations, one in North Africa and the other in a distillery in Cuba.

(b) Marine applications: A gas-generator, geared-turbine drive of 900 hp per shaft on a twin-shaft French mine sweeper. A single-shaft, 900-hp, geared-turbine installation on a newly constructed coastal vessel.

(c) Locomotive application[3, 4]: A single gas generator and turbine with a geared mechanical transmission (1000 shp). This locomotive application was accomplished in collaboration with La Regie Nationale Renault, the largest manufacturer of automobiles in France.

As a result of the success of the central stations installations, a 6000-kw (8 gas generators) plant is now on order. With respect to the marine application, some 23 gas-generator-turbine sets are on order or have been delivered for mine sweepers and commercial coastal vessels. All these installations incorporate single or twin units of the model GS-34 gas generator.

Some studies have been accomplished for a 2-generator 240-hp truck powerplant.[3, 4] A specific weight of less than 5 lb per shp appears feasible. It is believed, however, that because of the pressure of other commitments these studies have not progressed much beyond the drafting board.

The success of the French in the commercial exploitation of the free-piston-engine concept is the more spectacular when it is noted that it has been accomplished largely since 1946 in postwar France.

Switzerland – In this country the Sulzer organization constructed an experimental unit of approximately 2000 hp per generator.[5] This unit was of low speed (350 cpm) and great massiveness; and apparently their experience led them to abandon the project in 1946.

Germany – Junkers, of the famed engine manufacturing organization that bears his name, early recognized the potential of the free-piston-engine concept, especially since it would incorporate his 2-stroke opposed-piston-engine ideas. This interest, by 1935, led to the development of two models of air compressors, one a 70-cfm, 3000-psig, 4-stage unit (approximate engine rating 45 hp), and the

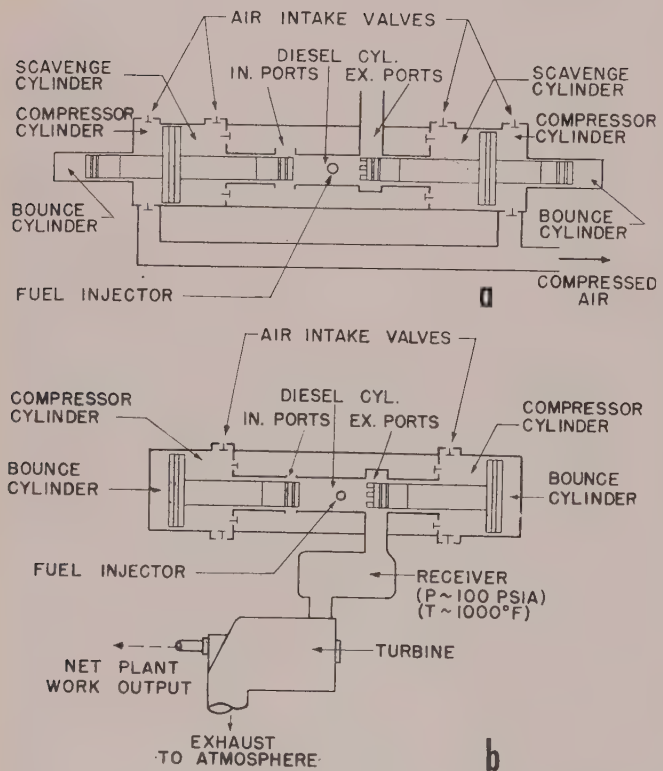


Fig. 1 – Basic applications of free-piston engine – (a) gas generator and turbine compound engine, (b) air compressor-diesel set

[3] See *Journal de la SIA*, Vol. 24, November, 1951, pp. 267-274: "Free-Piston Gas Generators" (in French), by R. Huber.

[4] See *Automotive Industries*, Vol. 106, Feb. 15, 1952, pp. 52, 106: "French Gas Turbine Truck Has Two Free-Piston Engines," by W. F. Bradley; Oct. 15, 1952, pp. 74, 90: "Free-Piston Engine and Gas Turbine Power French Experimental Locomotive," by W. F. Bradley.

[5] See *Engineer*, Vol. 181, May 10, 1946, pp. 428-429; May 17, pp. 460-461, May 24, pp. 483-485; May 31, pp. 506-507; June 7, pp. 525-526: "Research on Internal-Combustion Prime Movers," by J. Calderwood. (See especially pp. 484, 506.)

second a 380-cfm, 120-psig, 2-stage unit (approximate engine rating 100 hp).[6]

The high-pressure unit was early adopted by the German Admiralty as their standard shipboard unit for providing compressed air for torpedo launching. This system was more compact and weighed only about 35% as much as the standard electric-drive compressor used in the U. S. Navy for this service during World War II. The Junkers organization is currently marketing a 100-psig, 120-cfm, 2-stage, dolly-mounted unit. This model 2FK115A is the forerunner of a series of four models ranging from 70 cfm, 2850 psig to 425 cfm, 57 psig. Four stages are used for the high-pressure and a single stage of compression for the low-pressure unit.

Apparently the demand for compressors by the German Navy prevented Junkers from making much progress in the development of the compound-engine free-piston application.

Japan – Shortly after Pearl Harbor, the Japanese started manufacturing an identical copy of the high-pressure Junkers compressor for Naval use.[7] These were used extensively in both ship and shore installations, as evidenced by the considerable number of units picked up by the Australian and United States armed forces.

England – Pescara in the early prewar years arranged a license agreement with the British Alan Muntz and Co., Ltd. The present result of this arrangement is the commercial availability of the model P-42 Pescara-Muntz compressor, which is a 100-cfm, 90-psig, 23-hp (equivalent) dolly-mounted unit. This compressor is quite similar to the SIGMA P-13 unit. The major difference is the use of a helical-spring starting device instead of the compressed air ("gas spring") starting method used in the model P-13. There is some indication that Acrow (Engineers) Ltd. has arranged for a Junkers license.

Apparently not too much has been accomplished on the prime-mover application in England.

United States – After acquiring a Junkers compressor in 1942, the U. S. Navy commenced studying the possibilities of the free-piston-engine concept. This study led to a development contract with the Baldwin-Lima-Hamilton Corp. (at that time the Hoover-Owens-Rentschler Division of the General Machinery Corp.)[8] This contract has led to the development of several gas-generator models, one of which, the model B, a 2-generator single-turbine unit of 1000 kw, has undergone extensive testing by the Navy.[9] A second design, the model DL, is a prototype of a shipboard unit suitable, for example, for a mine sweeper powerplant. All that can be said about this powerplant at this time is that its thermodynamic performance is commendable and it has demonstrated durability.

Prior to the entry of Baldwin-Lima-Hamilton in this field, the Pratt and Whitney people made an intense effort to develop an aircraft powerplant, starting in 1941. The rapid development of the aircraft turbojet, plus the need for limiting their development activities, resulted in the abandonment of the project in 1946.[10]

The Cooper-Bessemer Corp., possibly as a result of the encroachment of the gas turbine powerplant as a gas line transmission prime mover, moved into the free-piston and turbine compound engine development in 1947, and their progress has been remarkable.[11, 12] Much of the rapidity of the development must be attributed to Robert P. Ramsey and William Morain. These engineers have been associated with the free-piston-engine development since early in 1943, first with the Baldwin-Lima-Hamilton organization and later with Cooper-Bessemer. Together with Frank M. Lewis, who is a consultant for Baldwin-Lima-Hamilton, these men may be considered as the "engineering fathers" of the free-piston-engine development in this country.

More recently, the General Motors Research Division has engaged in a cooperative development program with SIGMA. With the investigative resources of this company, it is not unreasonable to expect a fairly rapid evaluation of the commercial possibilities of free-piston machines in this country.

To date, relatively little interest has been shown in the development of light-weight gas generators in the 100- to 200-hp range. Possibly, this reflects the well-established commercial positions of the diesel and gasoline engines in this power range. A point of greater difficulty to understand is the apparent absence of any strong commercial interest in the air compressor application in this country, in spite of the considerable educational effort the Navy made in providing captured Junkers units to many manufacturers.

System Characteristics

Relative to the conventional crank-type engines, the major advantages of the free-piston and turbine compound engine are: (1) Substantially greater mechanical simplicity, providing for both lower manufacturing costs and ease of maintenance; (2) flexibility in application and excellent torque-speed characteristics because of the "fluid-coupled" turbine drive; (3) vibration-free operation afforded by the opposed-piston arrangement, resulting in a sub-

[6] See *VDI Zeitschrift*, Vol. 79, Feb. 9, 1935, pp. 155-160: "Junkers Free-Piston Compressor" (in German), by K. Neumann.

[7] See *Mechanical Engineering*, Vol. 67, December, 1945, pp. 852-854: "Free-Piston Compressor."

[8] "Development of High-Output Free-Piston Gas Generators," by R. L. Lasley and F. M. Lewis, ASME paper No. 53-S-34, presented at meeting, April 28-30, 1953.

[9] See ASME Transactions, Vol. 76, January, 1954, pp. 1-13: "Performance of Free-Piston Gas Generators," by J. J. McMullen and W. G. Payne.

[10] See pp. 154-155 of "The Pratt and Whitney Aircraft Story." Pub. by Pratt and Whitney Aircraft, Hartford, Conn., 1950.

[11] See Business Week, April 25, 1953, pp. 101-102, +: "Generating More Power from Less Fuel."

[12] See ASME Transactions, Vol. 76, January, 1954, pp. 15-29: "Free-Piston Type of Gas Turbine Plant and Applications," by J. J. McMullen and R. P. Ramsey.

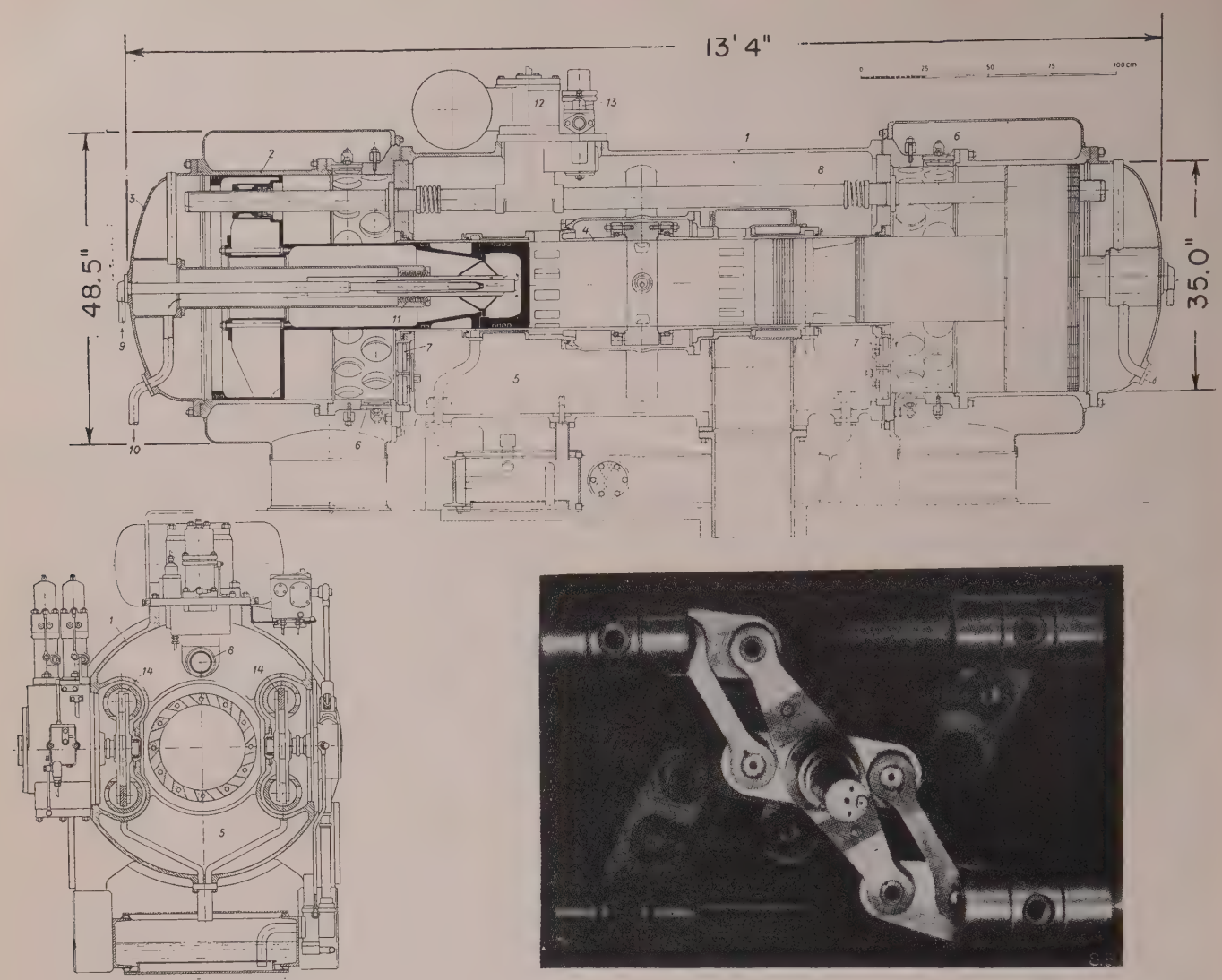


Fig. 2 – SIGMA type GS-34 free-piston gas generator – longitudinal elevation cross-section, transverse elevation cross-section, and double-exposure view of piston-synchronizing mechanism in both extreme positions. Specific weight 18 lb per shp, speed 618 cpm, rated at approximately 1000 turbine shp

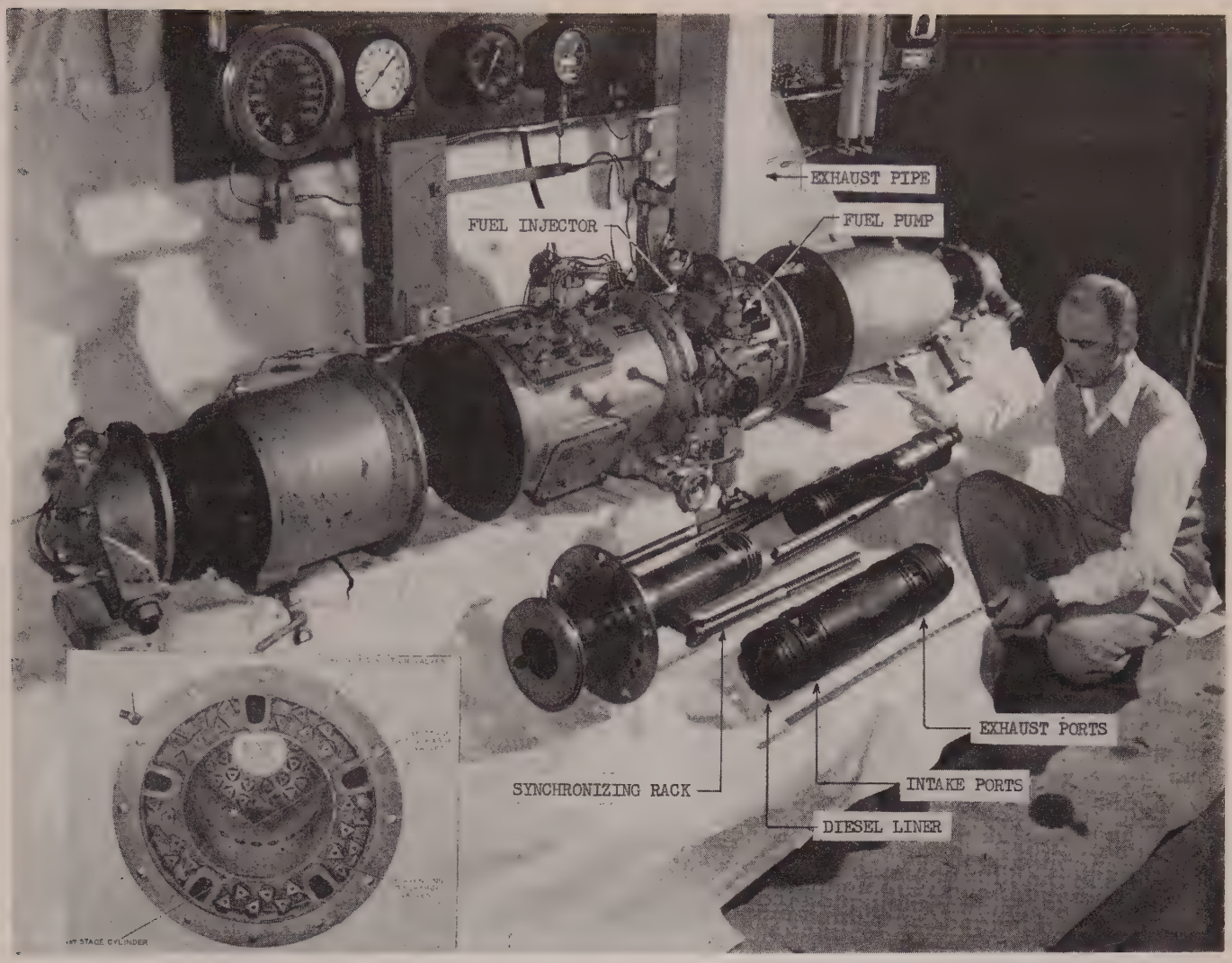
1 Casing
2 Compressor cylinder
3 Bounce-cylinder head
4 Engine cylinder
5 Scavenge chamber
6 Intake valves
7 Discharge valves
8 Pressure-equalizing pipe
9 Cooling water inlet
10 Cooling water outlet
11 Stuffing box
12 Starter mechanism
13 Stabilizer
14 Synchronizing mechanism

stantial economy in foundation requirements; (4) a relatively low specific weight. These advantages are gained with no sacrifice in fuel economy, as in this respect the free-piston compound engine compares favorably with the conventional diesel, the modern "record holder" for prime-mover thermal efficiency.

The free-piston and turbine compound engine has a better specific fuel consumption than even the most complex gas turbine cycle (intercooling, reheat, and regeneration). Moreover, this is achieved with a relatively low gas temperature to the turbine, obviating costly high-alloy construction.

Mechanical Features – The basic features of mechanical simplicity are illustrated in Fig. 2 (taken from an earlier paper[13]), which shows the details of the SIGMA model GS-34 unit. The performance of the first GS-34 reported by Eichelberg[2] is also given in another reference.[13] Current commercial

[13] See ASME Transactions, Vol. 74, November, 1952, pp. 1349-1361: "Free-Piston Engine Development – Present Status and Design Aspects," by A. L. London and A. K. Oppenheim.



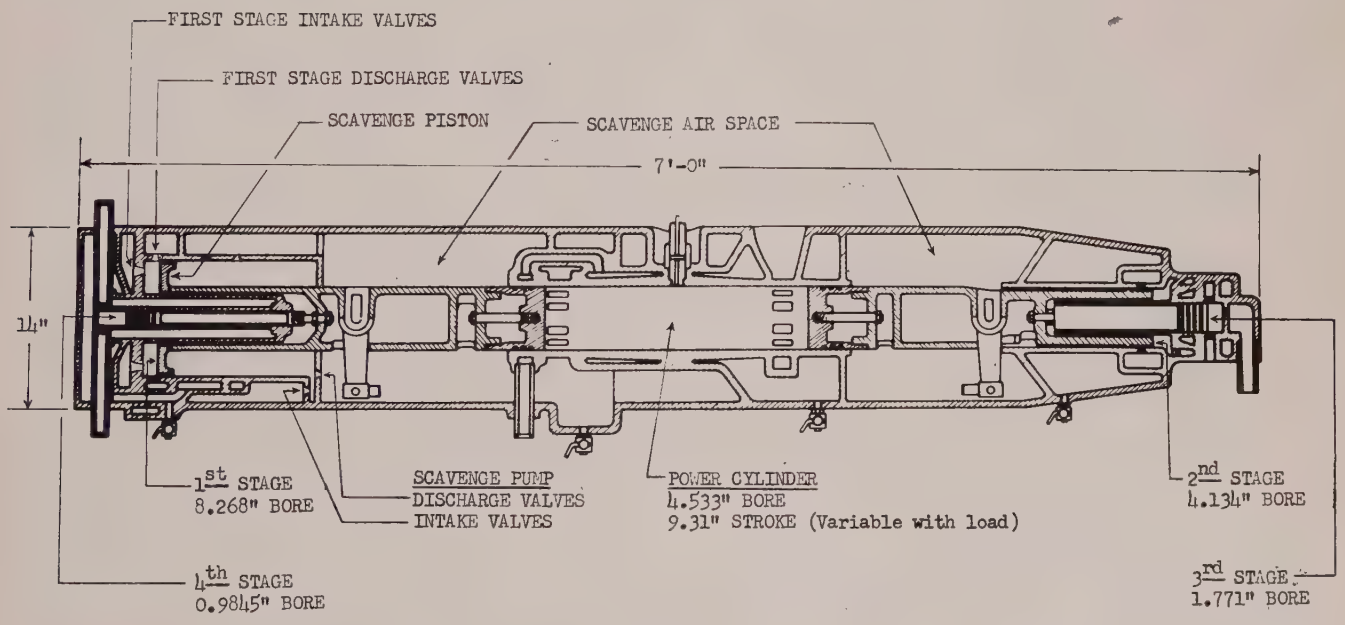


Fig. 3 – Junkers unit piston and cylinder arrangement – rated capacity 70.5 cfm free air, 2950 psig delivery, equivalent rating 45 shp

units have achieved a specific fuel consumption of 0.38-0.39 lb per shp-hr at full load. Fig. 3 (also taken from an earlier paper[13]) which shows the Junkers 4-stage compressor-diesel set, also highlights the feature of mechanical simplicity inherent in the free-piston-engine concept. Performance of the Junkers unit has already been reported in some detail.[13]

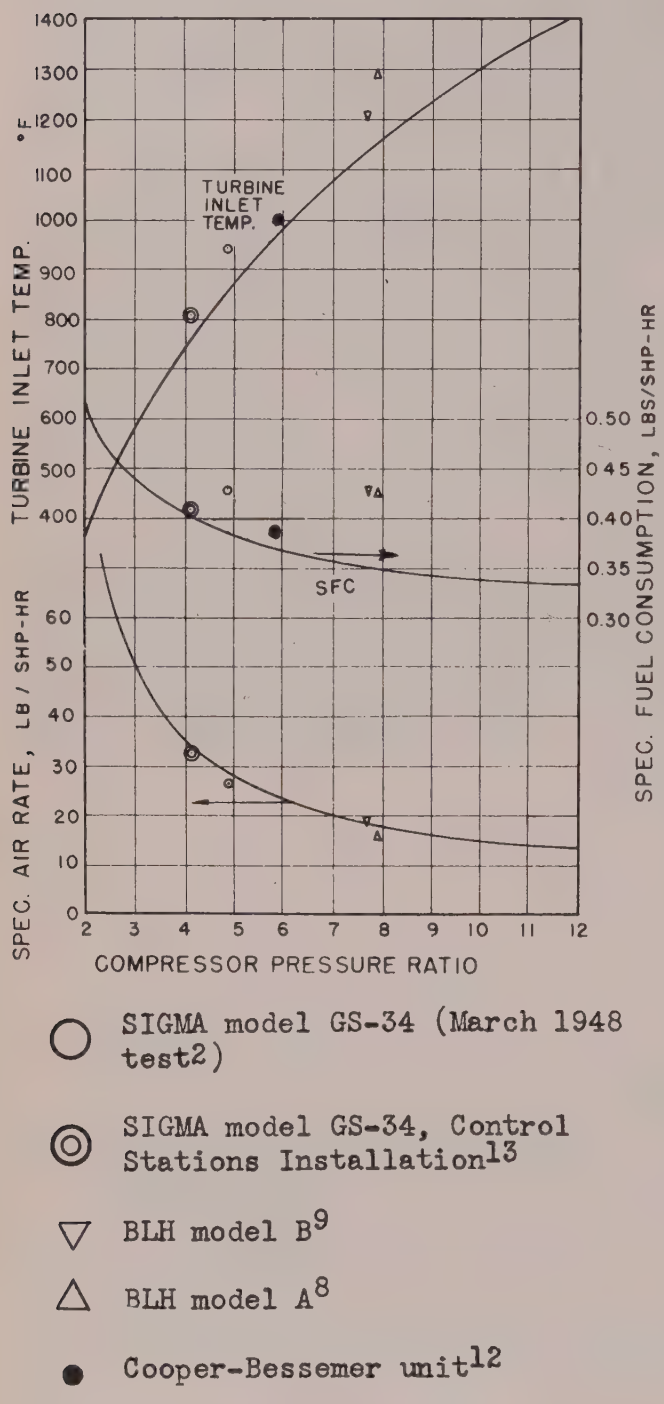


Fig. 4 – Predicted and test performance – see Table 1 for basis of prediction and Fig. 5 for additional predictions

Thermodynamic Considerations – Cycle studies involving only thermodynamic considerations can be readily accomplished by usual procedures. The results of such studies have already been reported.[14] These studies can be made to yield predictions for the following design parameters:

1. Specific flow rates in lb per shp-hr – turbine gas, engine fuel, diesel-engine air, and compressor air as functions of the supercharge pressure ratio.
2. Temperatures – engine charge and turbine inlet as functions of the supercharge pressure ratio.
3. Displacements – compressor cylinder relative to the engine cylinder for a specified compressor clearance and supercharge pressure ratio.
4. Engine indicated mep.

They would be based on estimates of:

1. Component efficiencies – compressor, turbine, engine mechanical, and engine indicated thermal.
2. Engine valve event pressure drops.
3. Engine fuelair ratio.
4. Coolant heat-transfer fraction.

Figs. 4 and 5 (both taken from another paper[14]) are examples of the results of such calculations. Superimposed are actual performance figures as obtained from the literature references indicated in the illustration. The lack of agreement between predicted and test performance is not too great when expressed on a percentage basis. The largest discrepancy occurs in the sfc figures for the two Baldwin-Lima-Hamilton units. When it is noted that these models were early developmental units and probably suffered from lower compressor efficiencies and larger engine valve losses than those estimated for the analysis, the 23% poorer sfc performance is understandable and, at the same time, indicative of the improvements that can be expected.

Table 1 was prepared from the results of the cycle analysis to indicate reasonable full-load performance expectations for a free-piston-type compound-engine system. It is to be noted that the specific fuel consumption is quite competitive with good crank-type diesel performance.

Other important information that can be deduced from a cycle analysis is the anticipated part-load sfc performance of the system. Such a result, based on a fixed-nozzle-area turbine and an assumed variation of turbine efficiency of 80% at full-load down to 67% at one-quarter load has already been reported.[13] It is summarized here in Fig. 6 (which is taken from another paper[14]). It is observed that relative to a crank-type diesel, the part-load econ-

[14] "Free-Piston and Turbine Compound Engine – Cycle Analysis," by A. L. London. ASME paper No. 53-A-212, presented at meeting, Nov. 29-Dec. 4, 1953.

Table 1 – Free-Piston and Turbine Compound Engine Predicted Performance

Gas Turbine	
Specific Flow Rate, lb per shp-hr	23.8
Gas Pressure, psig	65.5
Gas Temperature, F	970
Turbine Isentropic Efficiency,[a] %	80
Gas Generator	
Compressor:	
Pressure Ratio[a]	6
Displacement × Volumetric Efficiency/Effective Diesel Displacement	7.0
Engine:	
Volumetric Compression Ratio[a]	10.0
Combustion Air Fuel Ratio[a]	30
Engine Valve Events Pressure Drop Fraction,[a] Total $\Delta P/P$, %	10
Charge Temperature, F	506
Scavenger Air Combustion Air Ratio	1.13
Indicated Mean Effective Pressure, psi	334
Thermodynamic Data:	
Engine Indicated Thermal Efficiency, (lhv basis), %	41.1
Coolant Heat-Transfer Fraction[a], %	20
Mechanical Efficiency[a], %	90
Compressor Efficiency[a], %	80
Compound Engine System	
Specific Fuel Consumption[b]	0.30
Powerplant Thermal Efficiency (lhv basis), %	37.6
Powerplant Thermal Efficiency (hhv basis), %	35.1

[a]Denotes initially assumed factors.
[b]Based on a fuel lhv = 18,300 Btu per lb, hhv = 19,600 Btu per lb.

omy is not as good. A variable-area turbine nozzle design, or a reheat type of cycle as already described,[13] may improve this feature.

Thermodynamic-Dynamic Considerations – In Fig. 7 (which is taken from another paper[13]) the spring-mass nature of the free-piston engine is illustrated schematically. The speed of operation in cpm is determined by the mass of each of the two piston assemblies, which should be made equal, together with the character of the nonlinear gas springs.

The thermodynamic considerations will give, as previously shown, information on specific flow rates of gas, air, and fuel and the relative piston displacements of the compressor and engine cylinders. The design problem, however, requires a specification of absolute cylinder dimensions for a given power output. It is thus essential that the natural frequency or speed of operation be established; but, as pointed out, this requires a specification of piston mass that is directly related to the cylinder displacement, the quantity that it is desired to specify! This dilemma is resolved by making the thermodynamic-dynamic analysis on some sort of a "unit" basis, as for instance the "delivery of one pound of air per cylinder for each cycle." This temporarily fixes cylinder dimensions. The speed of operation is established next from successive numerical or graphical integrations of the differential equation of motion (Newton's second law), with the resultant force versus piston displacement characteristic established from the indicator cards for each cylinder, together with a specified allowance for friction forces.

In general the resulting turbine output will not

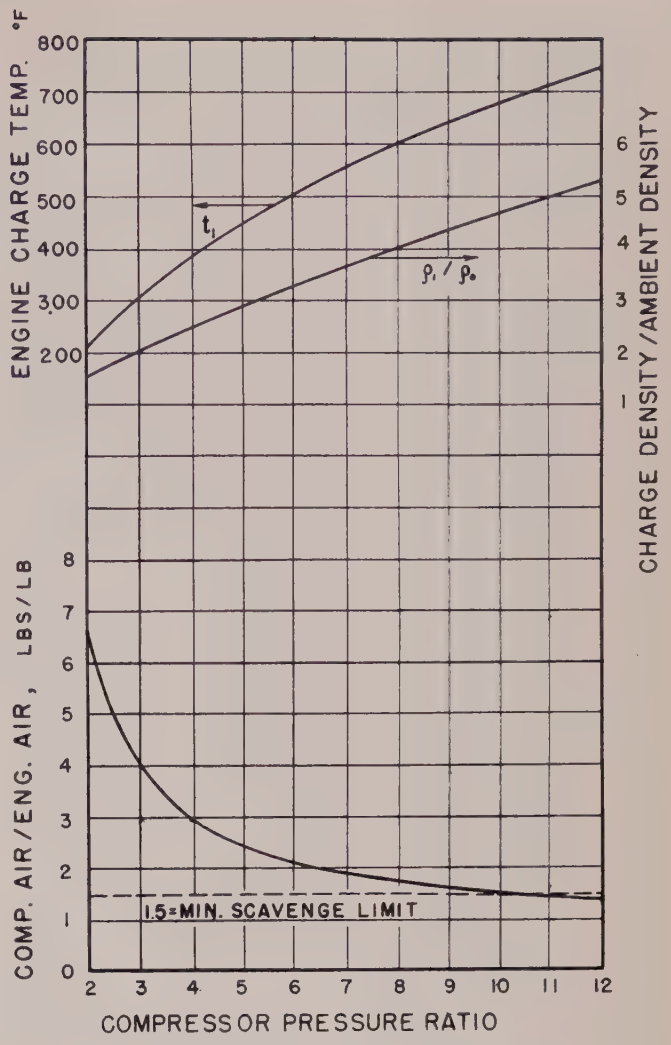


Fig. 5 – Predicted performance – continued from Fig. 4

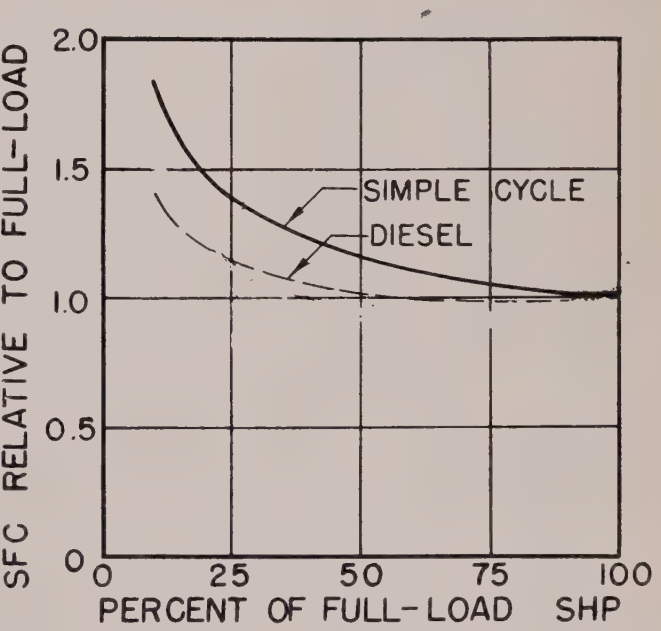


Fig. 6 – Part-load specific fuel consumption – based on fixed-nozzle-area type of turbine

correspond to the desired powerplant rating, but by the use of "affinity relations" it is possible to scale the gas-generator dimensions up or down to provide the necessary rate. Details of the thermodynamic-dynamic analysis and the development of the affinity relations have already been given.[13]

One of these affinity relations relates to the power rating

$$\text{hp} \propto D^2 \sqrt{s/(l\rho)} \qquad (1)$$

where D, s, l refer to diameter, stroke, and length dimensions respectively, and ρ refers to the "bulk" density of the piston mass (say 0.20 x 485 lb per cu ft for 20% "solidity" using steel). Another relation is derived for the frequency (f_r), in cpm:

$$f_r \propto 1/\sqrt{sl\rho} \qquad (2)$$

A third relation is given for specific weight in lb per shp:

$$\frac{M_o}{\text{hp}} \propto \frac{3}{l^2}\rho_o \sqrt{\rho/s} \qquad (3)$$

Here, ρ_o is the overall bulk density of the gas generator. These relations are presented here to illustrate how the affinity relations may be employed to scale from a given unit to another unit of a different rating. Suppose, as a simple example, we consider the 1000 shp SIGMA GS-34 unit shown in Fig. 2. It has a specific weight of about 18 lb per shp rating, and the dimensions are shown. What will be the dimensions, frequency of operation, and specific weight of a geometrically similar unit for a 100-shp rating application? For geometrical similarity, and for the same materials of construction, the dimensions s, D, l all vary in the same manner, and ρ is constant. As a consequence, equation (1) yields:

$$s, D, l \propto \sqrt{\text{hp}} = \sqrt{\frac{100}{1000}} = 1/3.16.$$

Thus the overall length of the gas generator would be 13.4/3.16 = 4.24 ft, the diameter 15.3 in., and the stroke 5.4 in. Equation (2) demonstrates that $f_r \propto 1/l$ so that the speed would be increased from 618 cpm for the SIGMA unit to 1950 cpm. The specific weight, from equation (3), would vary directly with the dimensions, and as a consequence be reduced from 18 lb per shp to 5.7 lb per shp, a quite acceptable figure, especially when it is realized that the SIGMA GS-34 is representative of heavy-duty design practices.

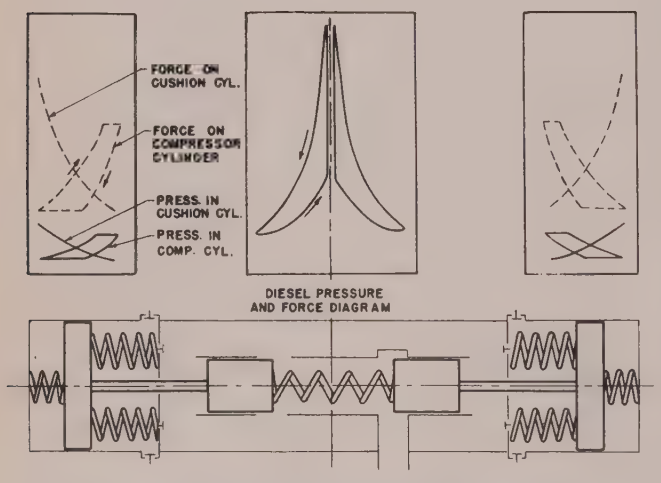

FORCE ON CUSHION CYL.
FORCE ON COMPRESSOR CYLINDER
PRESS. IN CUSHION CYL.
PRESS. IN COMP. CYL.
DIESEL PRESSURE AND FORCE DIAGRAM

Fig. 7 – Spring-mass nature of free-piston system – schematic illustration of relationship of indicator cards and nonlinear "gas springs"

The Future

It is believed that the French and German developments have already convincingly demonstrated the commercial feasibility of the compressor version of the free-piston-engine concept. In this application, however, the engine section operates at conventional 2-stroke diesel conditions and no unusual combustion and heat-transfer problems are involved. In contrast, the prime-mover version operates at high supercharge with the associated higher combustion rates per unit volume and higher charge density conditions, all accentuating the heat-transfer and ring and lubrication problems. Nevertheless, the moderate commercial success already achieved by the SIGMA model GS-34, the promising preliminary test performance of the Cooper-Bessemer unit, as alluded to by McMullen and Ramsey,[12] and the demonstrated durability of the Baldwin-Lima-Hamilton model DL unit, all suggest that the prime-mover system is rapidly becoming a proved unit of machinery. With the entry of General Motors and other organizations into the development, accelerated progress on this revolutionary engine concept in this country can be reasonably anticipated.

All the free-piston prime-mover applications that have been attempted are in the power range of 500-1500 hp per unit, and no serious efforts have been devoted to units in the 100-300-hp range. In view of the much greater ease and lower cost of development of a smaller unit it seems paradoxical that this is the case. However, both the Baldwin-Lima-Hamilton and SIGMA developments were Navy sponsored, and navies in general are more concerned with the development of larger marine propulsion units. In the case of Cooper-Bessemer, the development incentive was, apparently, the threat of competition of a 5000-hp gas turbine powerplant for gas line pumping, so that again circumstances forced the development of a unit of high power rating. In view of these circumstances, a reasonable expectation for the future is a greater

effort devoted to the units of lower power, possibly to compete with the heavy truck engine and the medium-speed diesels extensively employed in heavy-duty earth moving equipment.

In view of the commercial success already achieved in Europe with free-piston compressors, it may be that the conservatism toward this application encountered in this country will eventually be overcome, either through license agreements or through domestic developments. From a technical point of view the compressor set development possesses the attractive feature of providing a free-piston education free of the difficult heat-transfer and combustion problems of the prime-mover application. Moreover, because of the relative ease of the development, commercial production could be achieved in one or two years of effort.

The foregoing opinions are offered with the full realization that the patent situation has not been considered. All that will be noted here on this subject is that over 50 United States patents have been issued to the SIGMA-Pescara organization, about 16 to Junkers, and about 6 to the Sulzer organization of Switzerland.

Summary

The history of the development of the free-piston and turbine compound engine has been reviewed briefly. Salient mechanical and thermodynamic features of this revolutionary engine concept are considered and literature references containing more detailed treatments of these subjects are noted. The author's opinions as to the future of the development in this country have been aired, along with the customary qualifications.

As a concluding opinion, it is believed that early fruition of this most promising, but revolutionary, concept in engine design will require creative engineering talent of the first order. Failing this, sheer weight of effort, time, and the pressure of foreign successes will eventually result in progress here.

Acknowledgments

Since 1947 the Office of Naval Research and the Bureau of Ships have sponsored the free-piston-engine investigations conducted at Stanford University. The Bureau of Ships and the U. S. Naval Engineering Experiment Station furnished the Junkers diesel-compressor set pictured in Fig. 3 and the necessary replacement parts. Operating experience with this unit did much to provide the author and a number of Stanford engineering students with a practical education in free-piston machinery.

DISCUSSION

Foresees Better than 80% Efficiency for Turbine

– W. A. Morain
Cooper-Bessemer Corp.

OUR project was started early in 1950, rather than in 1947 as stated by the author. While the major development effort has been expended on a 1500-bhp, single-gas-generator turbine powerplant, other design studies have been worked out in some detail covering free-piston reciprocating compressors as well as turbine powerplants in the range from 100 to 12,000 bhp.

The values given in the author's Table I are in general quite conservative, and lead to a powerplant efficiency that is definitely in a competitive position with the present high-efficiency diesel engine. I believe we can expect much better than 80% efficiency from the turbine. Huber, in his discussion of a recent ASME paper, referred to a 6-stage turbine of 86% efficiency. The Cooper-Bessemer gas turbine is expected to give better than 84%, based on detailed calculations and design experience.

In the application of the affinity relationships in geometrically similar units, I would like to add a word of caution. There are limiting factors in wall thicknesses, passage areas, and permissible operating frequencies of such devices as valves, which must also be considered in scaling smaller units. Any wide variations in size should be carefully checked by design layout before settling on overall dimensions and weights.

With regard to the patent situation, it should be noted that many patents have been issued to American manufacturers of free-piston machinery and other applications are pending.

A discusser of this paper has made the statement that the free-piston combination could not achieve the efficiency of the diesel since they both had similar ring and friction losses, and the crank should be more efficient than the gas turbine. This is a popular misconception among those who have not taken the trouble to analyze the free-piston turbine cycle. A few words of explanation might be in order.

Let us assume that a crank-engine cylinder can develop, through suitable charging and porting, and so on, the same output as that of a free-piston cylinder. In the case of the C-B free-piston engine, this amounts to 200 psi mp at full load. It is difficult to visualize a 2-stroke crank-engine cylinder at this loading, but we will assume it can be done. With this cylinder loading, the C-B gas generator delivers 533 lb of gas per min at 1000 F and 72.5 psig to the gas turbine, when operating at 555 cpm. The gas turbine delivers 1500 bhp to the load at these gas-flow conditions. Referring the 1500 bhp back to the power cylinder displacement, we find it corresponds to 186 psi bmep. In other words, the power cylinder energy is converted to load at an efficiency of 92%. The crank engine cylinder, at a generally accepted mechanical efficiency of say 86%, would only develop 172 psi bmep, or about 1385 bhp through the crank mechanism. This is also confirmed in the overall by the exhaust temperature of the gas turbine, which is only about 550 F at this load rating, and which would run considerably higher on the crank engine, depending, of course, on the overall air/fuel ratio.

The free-piston gas turbine, as discussed in this article and those currently under test, must still be regarded as naturally aspirated, since the compressor takes its suction from the surrounding atmosphere. Even under these con-

ditions, the economic picture is a promising one, since estimated production costs for the overall free-piston turbine powerplants are below those of comparable turbocharged engines. Also, because of the complete absence of free forces or couples, the installation costs will be lower; there is no need for the huge blocks of concrete that the larger crank engines require for foundations. The free-piston turbine powerplant can be turbocharged readily, and will show encouraging power gain for even a mild boost. Through proper intercooling between the supercharger and free-piston compressor, the temperature levels through the cycle need not be raised appreciably. As I see it, there is no harm in letting the pressure levels rise, as long as rates of rise are controlled. If test observations eventually dictate certain pressure limits, the inner end clearance in the free-piston power cylinder can be increased to suit. Experience with conventional engines have already shown that, by maintaining the proper air/fuel ratio, there appears to be no definite limit in the permissible cylinder loading. This is especially true when one is not bothered with wristpin and crank bearings. It should be possible with proper attention to valve areas, flow paths, and combustion details to turbosupercharge the free-piston gas generator to at least double the naturally aspirated output. Overall efficiency will not be affected appreciably.

The many advantages of the free-piston gas turbine cycle are being confirmed by the manufacturers now engaged in developing this type of powerplant. The economics of weight, space, and cost are easily recognized. Fuel economies in line with those shown in Fig. 4 are being obtained. But there are still a few problems remaining to be completely solved, and only a relatively small number of engineers to work out these solutions. We believe that the performance data that will be released in the near future will serve to arouse the interest of a substantially increased number of engineers, which will, in turn, bring about more rapid progress.

Better Sfc Depends on Higher Allowable Cylinder Pressure

– Warren G. Payne
U. S. Naval Engineering Experiment Station

I WANT to start off by saying that, despite any criticisms I may make, I believe that the free-piston gas-generator turbine powerplant has very good prospects for both marine and stationary powerplants and, possibly, for automotive use. I have had the opportunity to follow the American developments closely and to see the results of the European work. All of the manufacturers have had to go through the tedious work of perfecting their units, but all are rapidly developing commercial units. The large number of installations already underway or planned for the SIGMA powerplants shows that they have been accepted.

I do feel that the shaft horsepower fuel consumptions predicted by the author are rather optimistic for current goals. The Baldwin-Lima-Hamilton Co. has obtained better thermal efficiency on both its model B and model DL than any of the other companies have reported. Their efficiency is about 5% better than the author showed for them in his Fig. 4. Actual gas horsepower fuel consumption at 7.8 pressure ratio was 0.325, or, allowing for an 80% efficient turbine, as he has done, 0.406 lb per shp-hr. It is anticipated that a turbine with an 85% full-load efficiency will be available and a consumption of 0.394 lb per shp-hr, after reduction gear, will be obtained. I believe the superior results that he has shown for the other American unit are what is expected when Baldwin-Lima-Hamilton has completed its development, rather than current status.

One difficulty in bringing the fuel consumption down to a better figure has been the maximum cylinder pressure. Allow that pressure to rise and fuel consumption gets better, but piston compression ring troubles get worse. A 1000-hr ring is the best any manufacturer has done, even with restriction of peak pressures. Development of a piston and ring assembly that will permit pressures to go up higher, say 2500 to 2800 psi, and fuel consumptions will be down in line with the curve of the author.

All the manufacturers are now restricting their hot gas temperatures to 1200 F or less in order to use turbines that are simple and inexpensive. Letting this temperature rise also improves thermal efficiency.

There can be considerable future reduction in the present rather high weight per horsepower ratio when supercharging is applied to the gas-generator compressor. This hasn't yet been done and there is some difference of opinion in the several theoretical analyses that have been made. Generally, however, it can be anticipated that the use of a 2-atmosphere turbosupercharger, installed after the power turbine as a first-stage air compressor, with an intercooler between it and the gas-generator compressor, could double the gas output with about the same thermal efficiency.

The free-piston gas-generator turbine powerplant has had comparatively little effort put into its development so far. I believe that an increase in that effort, with several companies engaged, will soon result in a powerplant that is simple in construction, low in initial cost, flexible in operation, and with an overall thermal efficiency about the same as the better diesels.

Believes Free-Piston Engine Can't Compete with Diesel

– R. Miller
Nordberg Mfg. Co.

ECONOMICS dictates that a prime mover must have, first, lowest possible cost; second, simplicity of construction; third, highest possible efficiency in converting heat into work; and fourth, reliability and low maintenance cost.

As a power generator, the free-piston gas turbine combination compares unfavorably with the modern diesel engine. In the diesel engine heat is converted to work in the reciprocating motion of the pistons, which is then converted to rotating motion by the highly efficient connecting-rod and crank mechanism. This mechanism is certainly more efficient, is lower in cost, and has greater simplicity than the gas turbine, with its reduction gears performing the same function of converting heat energy into power on a rotating shaft.

The modern diesel engine operates at its full load at the highest thermal and pressure load permissible with reliable operation and since pistons, piston rings, and liners are the weakest links and are common to both prime movers, it follows that the free-piston engine for equal reliability is limited to the same thermal and pressure load as the diesel engine.

The mean effective pressure in a diesel engine varies at constant thermal load as the absolute compression pressure to the exponent 0.383, as the combustion pressure to exponent 0.276, and inversely as the absolute compression temperature to exponent 1.352. Therefore, if the temperature of the air supplied to the free-piston cylinder is to be 500 F, as mentioned, the mean effective pressure can only be about 50% of the mean effective pressure obtained in the diesel engine with 100 F manifold temperature.

There is some justification for the free-piston engine as a compressor wherein the supercharging air can be intercooled and the reciprocating motion of the power piston is transferred directly without losses (other than piston-ring friction) to the compressor pistons.

It is interesting to learn from the author that the Sulzer Freres Co. has abandoned its free-piston power-generator development. I can only add that I compliment them on having made, in my opinion, a very sound decision.

Matching the Characteristics of Helicopters and Shaft-Turbines

D. N. Meyers and Z. M. Ciolkosz,
Piasecki Helicopter Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1953.

IN the last few years, the advantages of using shaft-turbines in helicopters have been recognized and discussed on many occasions. Most of these advantages have been so obvious that there is no need to go into a lengthy discussion here. Only a brief review will be presented covering the whole field of the application of shaft-turbines to helicopters. A somewhat more detailed analysis is given of the weight-saving possibilities of shaft-turbine installations, which help to counteract the disadvantage of high fuel consumption.

On the other hand, practically no information has been presented so far dealing with helicopter flight characteristics and how they are related to shaft-turbine power characteristics. An endeavor is made here to discuss this question and to propose some methods of obtaining better matching characteristics between helicopters and shaft-turbines.

General Review of Advantages of Shaft-Turbines

Smooth Torque Output of Turbine Shaft – The torque pulsations that occur in the piston engine impose a serious fatigue condition on the helicopter transmission system unless remedied by the use of a flywheel or flexible torsional coupling. Shaft-turbines do not have these torque pulsations.

Compactness and Ease of Installation – Since nearly all single-engine powerplants are housed within the helicopter body, it is imperative to keep the space assigned to the engine to a minimum. This is especially true when the helicopter con-

A DETAILED analysis of the weight-saving possibilities of shaft-turbine installations on helicopters is presented in this paper, which won the 1953 Wright Brothers Medal for its authors. They point out that this weight saving helps to counteract the disadvantage of high fuel consumption.

They also discuss helicopter flight characteristics and how they are related to shaft-turbine power characteristics. Finally, they propose some methods of obtaining better matching characteristics between helicopters and shaft-turbines.

The Authors

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Z. M. CIOLKOSZ is chief development design engineer, Piasecki Helicopter Corp. He was educated in Poland and received his degree of Master of Engineering Sciences there. After working for 13 years in the Polish aircraft industry and 7 years in the English aircraft industry, Mr. Ciolkosz joined Piasecki.

figuration is such that the engine is close to the helicopter center of gravity, since this region is the most desirable space for disposable load. In some cases, such as single-phase or tandem-seating helicopters, width of a reciprocating engine dictates the minimum fuselage size. A shaft-turbine will permit a much smaller cross-section, which in turn will result in less drag in forward flight, partly compensating for the difference in specific fuel consumption.

Low Oil Consumption and Ease of Cooling – Oil consumption in shaft-turbines is extremely small. The volume of oil carried is rather determined by the circulation of oil required for heat dissipation. The power section normally does not even need an oil cooler, although the reduction gears do. Sufficient flow through the oil cooler can be supplied by an exhaust ejector, which, at the same time, can ventilate the engine shrouding, thus eliminating a fan and stator.

Short Warmup Periods – Little or no warmup is required since it is not necessary to warm large quantities of oil before take-off. The shaft-turbine helicopter can be put into the air much more quickly than one with a piston engine.

Compressed-Air Availability – Bleed air from the turbine compressor may be used to drive aircraft accessories and operate a low-pressure pneumatic system.

Less Inflammable and Cheaper Fuel – A wide variety of fuels may be used, including those of less inflammable characteristics.

Improved Comfort – The intensity of noise from a shaft-turbine is likely to be roughly the same as from a piston engine, or perhaps somewhat greater, and it will be much higher in pitch. The aggregate noise level in helicopters, however, is comprised of noise from the engine, rotor blades, gearboxes, and vibration of different structural elements. The smoothness of turbine operation should result in a decrease in structural noise, so that the overall noise level should be at least no worse than with a piston engine, and is likely to be better. Vibration of higher frequency and smaller amplitude permits elimination or at least simplification of shock mounts. Improved comfort to crew and passengers will result from smoothness of operation because of elimination of engine vibration, which contributes to low-frequency noise as well as physical discomfort.

Improved Relationship between Power and Rpm – The shaft-turbine is inherently a high-speed engine and is especially advantageous for long driveshafts. A weight saving is possible if the major portion of the helicopter drive system is designed to operate at this high speed, with the final speed reduction made close to the rotor. There is more flexibility in choosing the optimum rpm for transmitting the power from the engine to the rotor.

Flexibility of "Free" Turbine – Some shaft-turbines have been designed with the output shaft driven by a separate turbine, not mechanically connected to the compressor and its driving turbine. This type of shaft turbine is ideal for helicopters. Some advantages of the "free" design are:

1. Torque characteristics that permit the rotor to absorb take-off power at a lower rpm than the high rpm desired for cruising. This results in a higher rotor efficiency for take-off and vertical climb. The rotors can operate over a much wider rpm range than with a reciprocating engine while still maintaining a constant compressor speed. This is also an advantage at high altitude, where a higher-than-normal rotor rpm may be desired.

2. Vibration and rapid load changes are not transmitted back to the compressor, a condition normally leading to turbine stability.

3. The control system for the powerplant is simplified and is not greatly different from a jet engine with a fixed nozzle.

4. Turbine inlet temperature is independent of output speed and torque. Thus, the danger of overtemperature is almost eliminated.

5. Ease of starting – only the compressor and its turbine need be accelerated. With a "solid" turbine it is necessary to accelerate the entire rotating mass including the reduction gearbox. This is a particular disadvantage in cold weather, when the oil in the gearbox is jelly-like.

6. In some cases, a starting clutch may be eliminated, since there is no mechanical connection between the gas-producer section and the power section.

7. The "free" turbine greatly simplifies the problem of coupling two or more engines together, with

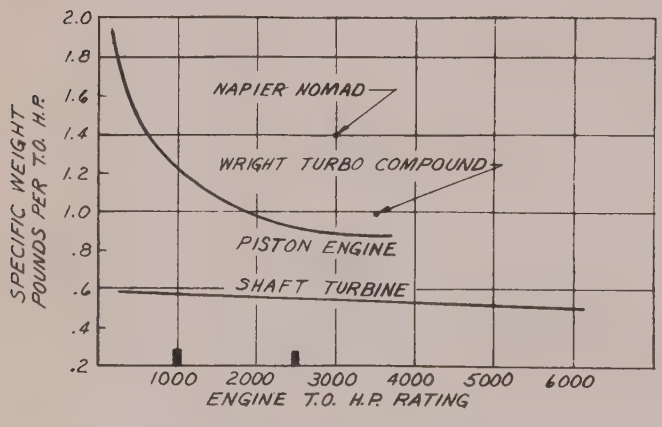

Fig. 1 – Specific weight versus engine size

better flexibility of the system in case of failure of one of the units.

Reduced Engine and Installation Weight – Although specific fuel consumption is still the greatest drawback to the economical application of shaft-turbines in helicopter design, a substantial reduction in engine installation weight offsets this disadvantage for short-range operations. While more fuel is consumed for the same range compared to a piston engine, the cost will be significantly influenced by using a cheaper fuel, much less oil, and by the ease of maintenance associated with simplicity of installation, improved accessibility, and less vibration.

Design Trends and Engine Development

Let us compare the development of piston engines and shaft turbines.

Specific Weight – Fig. 1 shows the trend in specific weight of piston engines and shaft-turbines. Specific weight is plotted on the basis of engine size, according to its take-off rating. The piston engine, which has made aviation history from its very beginning, has become very much stabilized. The curve shown, which is representative of the latest-production piston engines, is little different from a similar curve based on the latest engines of five years ago. Distinct departures from the general trend are the development of the Napier Nomad and the Wright Turbo Compound. These two engines both are partly piston engine, partly gas turbine, and have a higher specific weight than the general trend.

The trend of shaft-turbine specific weight is shown for comparison on the same graph. This trend is quite representative of present-day turbine development, based on turbines either in production or scheduled for production. In a few cases corrections were made to published turbine weights to account for such special details as a gearbox for counterrotating propeller shafts, which would have increased the specific weight of the turbine.

Since the specific weight of both turbines and piston engines, especially the latter, clearly decreases with size of engine, it is pointless to obtain any average figure embracing all sizes. However, the horsepower range can be conveniently divided into three categories: low power (less than 1000 hp), medium power (1000 to 2500 hp), and high power (more than 2500 hp). We can now say that in the small-power category, shaft-turbines weigh an average of 0.58 lb per hp or 40% of piston engines; in the medium-power category, shaft-turbines weigh 0.57 lb per hp or 56% of piston engines; in the high-power category, shaft-turbines weigh 0.54 lb per hp or 63% of piston engines.

Specific Fuel Consumption[1] – Fig. 2 shows the trend in specific fuel consumption for piston engines and shaft-turbines. Difficulty was encountered in obtaining data for these curves in that, in most cases, particularly for the gas turbines, fuel consumption was quoted only for take-off power. However, for those engines for which consumption data were available at normal rated power as well as take-off, it was found that there was a nearly constant ratio between the specific fuel consumption at normal rated power and that at take-off. This ratio for piston engines is, of course, different from gas turbines, but again was found to be nearly constant for all the piston engines for which data were available. In Fig. 2 the dash lines represent the trend in sfc at take-off power, taken directly from published data on the engines concerned. The solid lines represent the sfc at normal rated power as determined by the correction factor just described. It can be seen that the fuel consumption for piston engines improves with a reduction in power, while that for gas turbines becomes worse. Furthermore, note that the fuel consumption for the large-size turbines is much better than for the small ones (just the opposite from the piston-engine trend). Once again the Napier Nomad and the Wright Turbo Compound are exceptions to the general trend, with significant improvement in fuel consumption. For the small piston engine, there is little difference in consumption between normal rated power and cruise power. For the large engines, however, cruise fuel consumption is roughly half of the consumption at normal rated power.

Since take-off power of an engine is restricted to 5-min operation, fuel consumption at take-off power is of little significance. For this reason, any comparison between shaft-turbine and piston engine should be made at maximum continuous, or normal rated, power. Comparing the fuel consump-

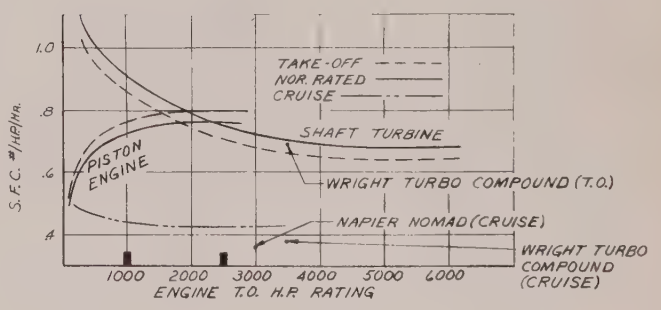

Fig. 2 – Specific fuel consumption versus engine size

[1] Specific fuel consumption is based on shaft horsepower (jet thrust not included) in this discussion.

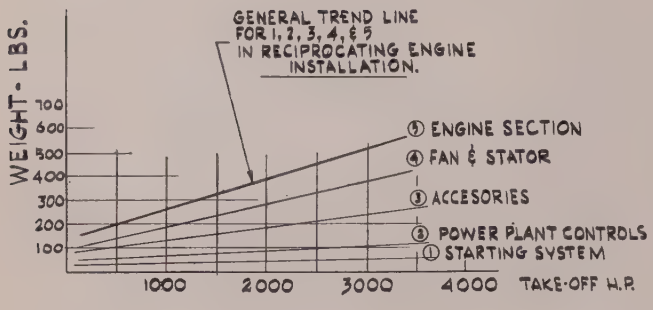

Fig. 3 – Engine section and accessories weight for reciprocating engines

tion curves, it is seen that, for powerplants of more than 2500 hp, the shaft-turbine is already superior to the piston engine at normal rated power. This superiority will probably increase in the next few years since we can expect further improvement in shaft-turbines but little, if any, improvement in piston engines. There is a clear application, then, for shaft-turbines whenever normal operation demands power approaching the maximum continuous rating. The piston engine, however, is apt to be superior in applications requiring cruising at reduced power for long periods of time.

Factors Involved in Better Turbine Performance – A logical question now arises: What is being done to reduce specific fuel consumption and increase specific output (hp per lb of weight) of gas turbines? These two improvements are, to some extent, inseparable, and both will help make shaft-

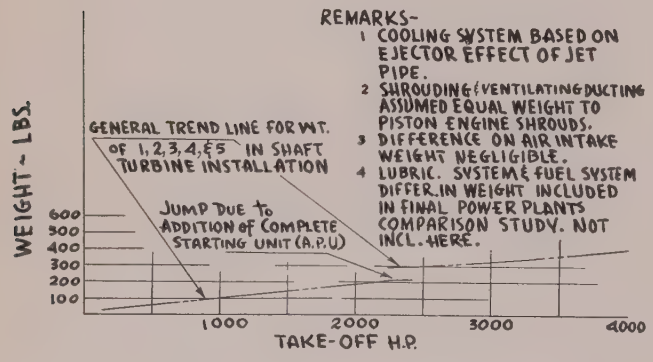
REMARKS-
1 COOLING SYSTEM BASED ON EJECTOR EFFECT OF JET PIPE.
2 SHROUDING & VENTILATING DUCTING ASSUMED EQUAL WEIGHT TO PISTON ENGINE SHROUDS.
3 DIFFERENCE ON AIR INTAKE WEIGHT NEGLIGIBLE.
4 LUBRIC. SYSTEM & FUEL SYSTEM DIFFER. IN WEIGHT INCLUDED IN FINAL POWER PLANTS COMPARISON STUDY. NOT INCL. HERE.
GENERAL TREND LINE FOR WT. OF 1, 2, 3, 4, & 5 IN SHAFT TURBINE INSTALLATION
JUMP DUE to ADDITION of COMPLETE STARTING UNIT (A.P.U.)
WEIGHT - LBS.
600
500
400
300
200
100
1000
2000
3000
4000
TAKE-OFF H.P.

Fig. 4 – Engine section and accessories weight for shaft-turbines

turbines more attractive and competitive for use in helicopters. Fortunately, work on both improvements is being conducted continuously and information coming from the engine industry is most encouraging.

The open-cycle continuous-flow gas turbine does not depend, like the internal-combustion engine, on the development of better fuels for improved efficiency. Its progress is associated with advancement in mechanical and metallurgical engineering.

The improvements of a mechanical nature are chiefly in efficiency of the compressor and the turbine. Although compressor and turbine efficiencies above 0.85 are already attainable, even small improvements in these efficiencies are very important to overall thermal efficiency and net power output. Only the excess of turbine shaft power available over the requirements of the compressor becomes useful power at the propeller shaft. Approximately two-thirds of the power developed by the turbine is used to drive the compressor, and only one-third is available for useful work. For the same fuel consumption, therefore, a 2% improvement in turbine efficiency increases the net output by 6%, and a 2% improvement in compressor efficiency increases the net output by 4%.

Metallurgical advances involve the ability to use high turbine inlet temperatures through the use of improved alloys. Higher turbine inlet temperature represents an effective means of improving the thermal efficiency of the powerplant and increasing the net output. Therefore, it has considerable importance on the weight and size of the powerplant and its specific fuel consumption.

On the other hand, to take full advantage of the increase in turbine inlet temperature, higher pressure ratios are required, which tend to make the compressor longer and heavier. Considerable improvement in this field is in progress, however, mostly in the increase in pressure ratio per stage. This is being done by continuous research and by development in transonic and supersonic compressors. An increase in pressure ratio per stage has a beneficial effect in three ways:

1. It permits a higher overall pressure ratio without extending the length of the engine.

2. For a given pressure ratio, the number of stages is reduced, with a resulting decrease in weight, since the compressor usually weighs 25 to 30% of the engine weight.

3. It alleviates the complexity of a multistage design with a large number of stages.

In general, the above-mentioned factors, such as compressor efficiency, turbine efficiency, turbine inlet temperature, and pressure ratio, are all interdependent in different degrees. All these factors are involved in the thermal efficiency of the powerplant, its specific fuel consumption, its maximum power output, and its specific weight. Small gains

may also be obtained in shaft power output and fuel consumption by cutting down to the minimum the energy represented by the exhaust gases as jet thrust. The elimination of this thrust is of further advantage, since it must otherwise be compensated in helicopter design.

Since so many variables are involved in the determination of optimum efficiency and maximum output of shaft-turbines, predictions in trends are difficult to make. The improvements will depend on the nature of the problems attacked by different engine companies and on the results of their solutions. Generally, it may be stated, however, that reduction of specific weight will progress more rapidly than reduction in specific fuel consumption. In some cases, in the small power bracket, engine companies are promising reductions in specific weight and specific fuel consumption well below present average figures.

Effect on Helicopter Payload – In Fig. 1, it has already been shown that the basic powerplant weight is considerably less for a shaft-turbine than for a piston engine of the same hp. The basic engine weight, however, is not the whole story. To complete the investigation on installed powerplant weight saving, four more graphs are presented. Fig. 3 shows the trend in installed weight of the engine section, cooling system, accessories, powerplant control, and starting system for piston engines plotted against take-off hp of the engine. Fig. 4 shows the trend in weights for these same items of engine installation as applied to shaft-turbines, less the cooling system, which may usually be eliminated from shaft-turbine designs. The discontinuity in the trend line of Fig. 4 is because an auxiliary power unit becomes necessary to start gas turbines of sizes in excess of 2000 hp. Fig. 5 shows the trend of Figs. 3 and 4 superimposed. The shaded area represents weight saving from the use of shaft-turbines over and above the difference in weight of the basic powerplant. This information, combined with the weight saving from the basic engine (taken from Fig. 1) will show how much the weight empty of the helicopter can be reduced by using a shaft-turbine.

The influence of this weight saving is shown in Fig. 6, which gives the total of installed powerplant weight and fuel weight (for any given range) plotted against range. These plots are for hypothetical helicopters, all of the same configuration and size, differing only in the type of powerplant. Take-off hp is the same in both cases. The two horizontal base lines represent the total installed weight of a piston engine and a shaft-turbine, respectively, including all necessary accessories. Weight allowance for fuel tank was made on the basis of the amount of fuel required for each range. Since the gas turbine has a higher fuel consumption than the piston engine, the line representing the fuel tank weight for the gas turbine is of a somewhat higher slope than for the piston engine. The difference in fuel

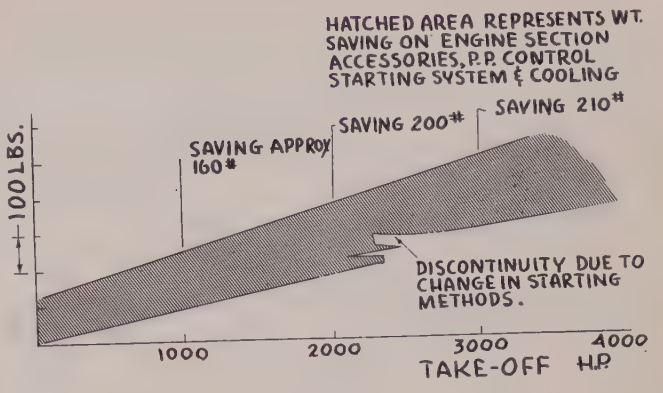

Fig. 5 – Difference in engine section and accessories weight between shaft-turbines and reciprocating engines

consumption itself is shown in the difference in slopes of the lines representing total fuel weight. It should be pointed out that the piston-engine helicopter was considered to operate at its speed for best range, which is well below its maximum speed. The shaft-turbine helicopter, however, was considered to operate at its maximum speed, which is also its speed for best range.

Point A, where the lines representing fuel weight intersect, shows the range at which a helicopter

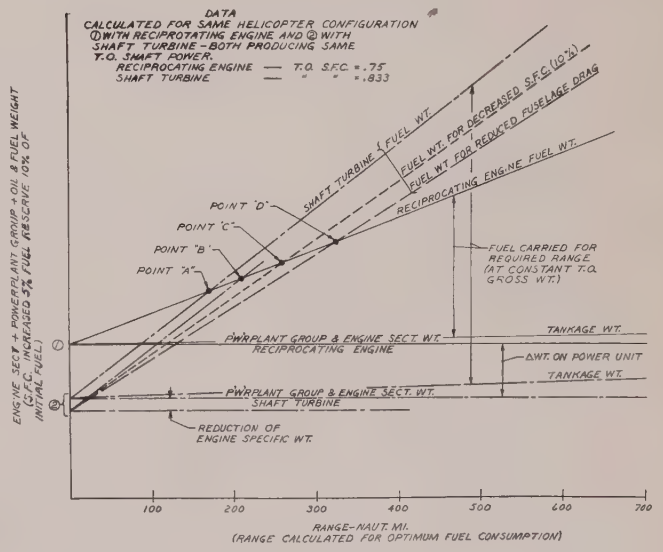
DATA
CALCULATED FOR SAME HELICOPTER CONFIGURATION
① WITH RECIPROTATING ENGINE AND ② WITH SHAFT TURBINE – BOTH PRODUCING SAME T.O. SHAFT POWER.
RECIPROCATING ENGINE — T.O. S.F.C. = .75
SHAFT TURBINE — " " = .833
SHAFT TURBINE & FUEL WT.
FUEL WT. FOR DECREASED S.F.C. (10%)
FUEL WT. FOR REDUCED FUSELAGE DRAG
RECIPROCATING ENGINE FUEL WT.
POINT "D"
POINT "C"
POINT "B"
POINT "A"
FUEL CARRIED FOR REQUIRED RANGE (AT CONSTANT T.O. GROSS WT.)
TANKAGE WT.
PWRPLANT GROUP & ENGINE SECT. WT. RECIPROCATING ENGINE
Δ WT. ON POWER UNIT
TANKAGE WT.
PWRPLANT GROUP & ENGINE SECT. WT. SHAFT TURBINE
REDUCTION OF ENGINE SPECIFIC WT.
ENGINE SECT + POWERPLANT GROUP + OIL & FUEL WEIGHT (FUEL INCREASED 5% FUEL RESERVE 10% OF INITIAL FUEL)
100
200
300
400
500
600
700
RANGE-NAUT. MI.
(RANGE CALCULATED FOR OPTIMUM FUEL CONSUMPTION)

Fig. 6 – Range comparison for helicopters with reciprocating engines and with shaft turbines

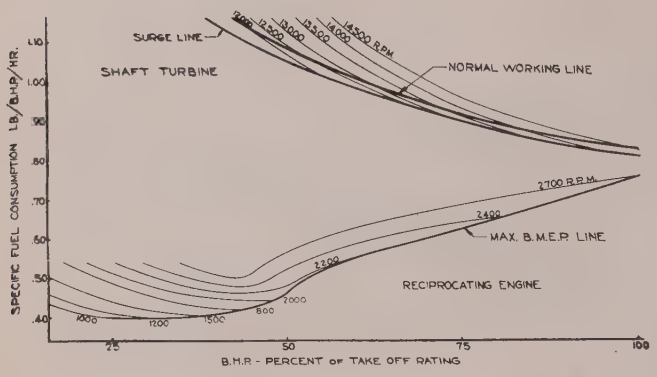

Fig. 7 – Typical engine performance at sea level

with either a shaft-turbine or a reciprocating engine would have the same payload. A shaft-turbine was intentionally chosen with a rather high specific weight (0.61 lb per hp) for this example, but with the possibility of future improvements in specific weight and specific fuel consumption, which could be expected within a few years. It is assumed that the specific weight may be expected to be reduced to 0.58 lb per hp. This reduced engine weight is shown as a downward shift in the base line representing installed weight of the shaft-turbine. The range at which the payload becomes equal to the piston engine is thus increased more than 20% to point B. A reduction in sfc of 10% would amount to a further increase in range of about 50% of the initial value to point C. A further possibility of improvement is a reduction of fuselage cross-section and parasite drag. As mentioned earlier, piston-engine installation in the fuselage sometimes dictates a bigger fuselage cross-section than would be necessary otherwise. The possible decrease in drag by means of cutting down the cross-sectional area results in the intersection point B, which further increases the range quite considerably.

It is appropriate at this point to emphasize strongly the importance of drag reduction to a helicopter with a shaft-turbine. Since, for good range, flight will normally be executed at a cruising speed close to V_{max}, where parasite drag is high, reduction of drag for cruising flight, due to clean design, will be much more beneficial than in a helicopter with a piston engine.

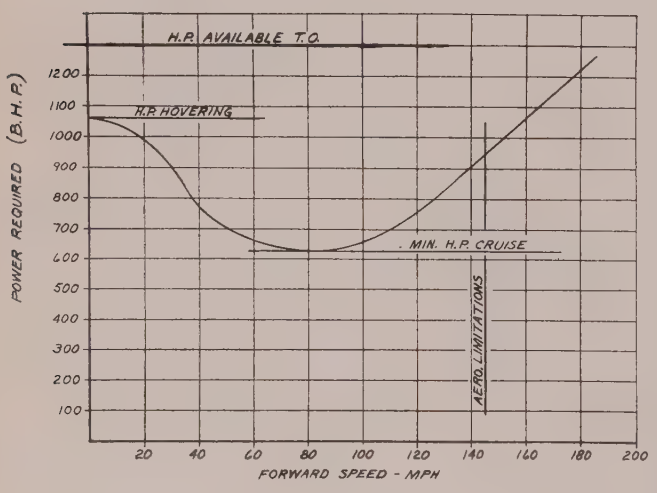

Fig. 8 – Helicopter power required in forward flight at sea level

Helicopters With Turbine and With Piston Engines

The difference in operating characteristics between shaft-turbines and piston engines is responsible for some performance differences between helicopters powered by these two different powerplants. These performance characteristics, in turn, make the shaft-turbine well suited to a number of helicopter operating conditions, and make it unsuited to others.

Discussion of Payload, Range, and Speed – It is in order first to compare the fuel consumption characteristics of a piston engine with those of a shaft-turbine. Fig. 7 shows this comparison for a typical medium-power piston engine and shaft-turbine. It is quite obvious how the sfc of the piston engine continuously decreases with a decrease in power output, particularly if the best engine rpm is chosen for each power output.

The characteristics of the shaft-turbine, however, are exactly the opposite. The sfc continuously increases as the power output is reduced. Furthermore, although the power output that can be obtained from a shaft-turbine is somewhat independent of rpm, this is true only over a rather narrow range. This rpm range is limited by the danger of overheating the turbine and surging of the compressor. The shaft-turbine operational range is usually enclosed between the so-called "normal working line" and the line representing maximum permissible rpm. In helicopter operations, the working range is further limited between the maximum permissible rpm and the rpm corresponding to rotor blade stall.

It is obvious from Fig. 7 that, in order to obtain maximum range, a shaft-turbine helicopter must fly at approximately its maximum speed, while the piston-engine helicopter must fly at some reduced speed and reduced power setting. This is further illustrated in Fig. 8, which shows the power required for a typical helicopter to fly at various speeds. The maximum speed is shown as cut off by aerodynamic limitations, such as stalling of the

retreating blade or compressibility effects on the advancing blade. The upper horizontal line represents the installed available hp and the vertical distance between this line and the power-required curve represents the excess power available for climb at any speed. This excess power would also normally allow for adequate performance at high altitude or in very hot weather, since the power-required curve represents the standard sea-level conditions.

Applying the sfc data from the previous Fig. 7 to this hypothetical helicopter results in Fig. 9, which shows miles flown per lb of fuel expended as a function of forward speed. The rpm associated with each forward speed and power required has been chosen here to fit the helicopter characteristics. This rpm is not necessarily the one that gives the lowest sfc when the engine alone is considered. As mentioned before, the highest value of miles per lb, and therefore the greatest range, occurs at maximum speed with a shaft-turbine, but at a considerably lower speed with a piston engine.

Helicopter Applications for Which the Shaft-Turbine is Suited – An analysis of Fig. 6 and Fig. 9 shows that the shaft-turbine is competitive to the piston engine, for use in helicopters for the following applications:

1. In transport operation, right now, for ranges up to about 200 miles, with the strong possibility of increasing this competitive range in the near future to about 350 miles. To keep sfc to a minimum, flight should be executed at high speed, close to V_{max}. This is really an advantage since speed is normally the prime objective in flying.

2. "Crane" or short ferry type of missions in which most of the time is spent in hovering or extremely slow forward speeds when the power required is high, and when the sfc of the turbine approaches that of the piston engine.

3. Operation in low-temperature areas, such as arctic or subarctic regions. Since the efficiency of a turbine powerplant at a fixed pressure ratio depends upon the temperature at the compressor inlet, a decrease in this temperature will considerably improve the thermal efficiency and consequently reduce sfc. The net output will also be increased because of the reduction in compressor work.

4. Missions that, by their nature, are required to be flown at high cruising speeds, such as urgent rescue missions.

5. Transportation of freight or passengers between points a short distance apart, but of greatly different elevation, where most of the cruise time is spent in climbing.

Helicopter Applications for Which Present Turbine Characteristics are Unsuited – These are:

1. Cruise at minimum power. The optimum cruising speed for a shaft-turbine helicopter is its maximum speed. Any mission associated with an extended flight at minimum power requirements will be rather inefficient, and the range and endurance will be less than for a helicopter with a piston engine. Some methods for improving this condition are discussed in a following section of this paper.

2. Missions in which emphasis is placed upon endurance time rather than range.

3. When speed is limited by aerodynamic limitations, such as stall or compressibility, to a value well below the speed that otherwise could be obtained by the power available. Such a limitation will compel flight at a speed that does not correspond to the optimum cruising speed and will, therefore, result in a low value of miles flown per lb of fuel.

4. Missions that require most of the flight to be in high-temperature areas. As the temperature at the compressor inlet is increased, thermal efficiency and net power output will both be reduced. This will require installation of a larger engine, operating at a reduced rating, which in turn will result in a reduction in payload.

5. Carrying equipment or freight outside the fuselage for long-distance flight. Parasite drag will be much more serious for a turbine helicopter because it is normally flying at a higher speed.

Adapting Helicopter Characteristics to Shaft-Turbine

In the previous section, a number of helicopter operations were pointed out to which the shaft-turbine is not well adapted under conventional operating techniques. There are methods of overcoming

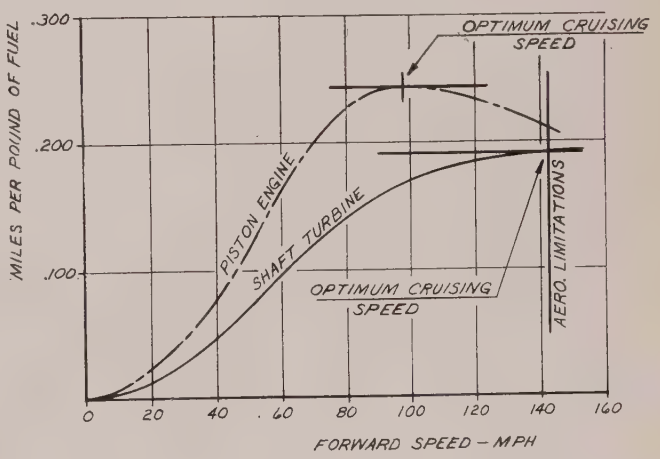

Fig. 9 – Miles per pound of fuel versus forward speed at sea level

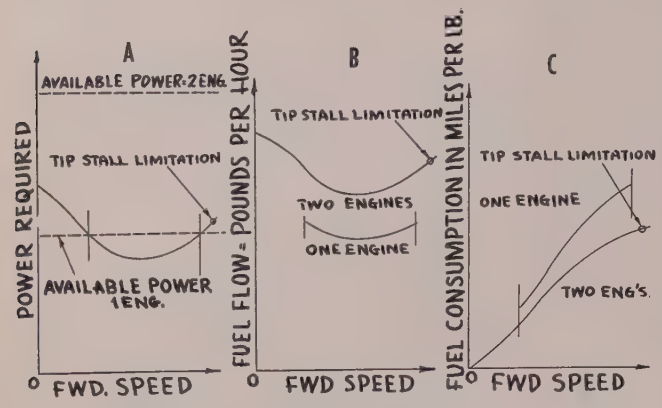

Fig. 10 – Multiengine helicopter performance

these deficiencies, however, as will be pointed out here.

Multiengine Design – The shaft-turbine characteristic that it is most important to match in the design of the helicopter is the variation of fuel consumption with per cent of rated power. All shaft-turbines now operating or under design have the common feature of rapidly increasing specific fuel consumption with decrease in power output. A typical fuel consumption curve was shown in Fig. 7. By using two or more small shaft-turbines instead of one large one, the helicopter is enabled to fly with one turbine shut down and the other operating at or near maximum continuous power and, therefore, at nearly best possible fuel consumption. This is illustrated in Fig. 10. Graph A shows the power required for a typical twin-engine helicopter as a function of forward speed. The two horizontal lines show the power available from each of the two turbines. It is apparent that, throughout the entire speed range of this helicopter, the power required is considerably less than the combined power of both turbines. There is, however, a moderate-speed range within which the helicopter can fly on one turbine. Curve B shows the fuel flow of this helicopter flying with both engines operating, as well as with one engine shut off, and curve C shows the miles flown per pound of fuel under these same conditions.

A shaft-turbine can be made to deliver full power in a fraction of a minute after a cold start. No warmup is required, as in the case of a piston engine. For this reason, it is feasible to operate a multiturbine aircraft with some of the turbines shut down. A typical shaft-turbine can be brought from a cold start to full power in 30 sec. If a twin-turbine helicopter, therefore, operating on one turbine should experience powerplant failure, the second turbine could be operating by the time the helicopter had lost 800 ft of altitude. It would, of course, be unsafe to operate thus with piston engines because, in case of emergency, the reserve engine would be cold and could not be depended upon. It is also interesting to note that no benefit would be obtained from operating piston engines with some of them shut down, since these engines always have their best fuel consumption at part throttle rather than at full power.

High-Speed Design Features – The previous section discussed methods for permitting economical shaft-turbine operation at moderate cruising speeds by using smaller power units operating at maximum continuous power. Another approach is to design the helicopter so that it can fly at a speed fast enough to absorb the full installed horsepower. In general, modern conventional helicopters having sufficient power for good hovering performance are unable to use this power in forward flight. V_{max} is usually limited by aerodynamic considerations rather than by power. The usual limit is tip stall of the retreating blade. An attempt to fly faster than this limit results in extreme vibration even though the power required may be well below the available horsepower.

Use of Wings and Propellers – Fig. 11A shows the limiting tip-speed ratio (μ_s) plotted against rotor tilt angle (γ) for different values of blade loading (C_T). It can be seen that, if the rotor is tilted further forward to obtain propulsive thrust, the limiting tip-speed ratio decreases. This limiting tip-speed ratio is further decreased by an increase in blade loading. This graph is completely nondimensional. Graph B has been derived from graph A using values typical of current helicopters and shows V_{max} in miles per hour as limited by tip stall for different rotor tip-speeds. The dash line on this

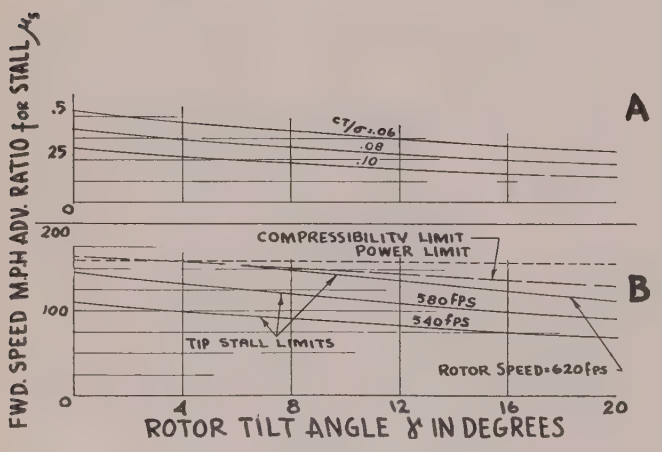

Fig. 11 – Limitations to helicopter forward speed

graph represents the V_{max} as limited by installed horsepower.

For a particular helicopter of the conventional type, the value of rotor tilt is fixed for each forward speed since it is the horizontal component of this forward-tilted thrust vector that provides the propulsive force to overcome drag. It can be seen from Fig. 11B that the maximum forward speed can be increased by means of either an increase in rotor tip-speed or a decrease in tilt angle. There is a limit to the permissible increase in rotor tip-speed, however, since eventually the advancing blade approaches the speed of sound and compressibility effects begin to cause vibration. This is shown by the compressibility limit in Fig. 11B. Thus, after the tip-speed is increased as much as possible, there remains only the method of decreasing the rotor tilt angle. The way to accomplish this is to provide the forward propulsion by means of a propeller instead of by tilting the rotor forward. In this way, the aircraft can be designed to fly efficiently at a high cruising speed, using the full continuous power of the shaft-turbine, which then will operate at better fuel consumption.

It is possible that, with reasonable rotor parameters, tip stall might be reached at a speed that still does not require full power, even with the use of a propeller. In such a case, a fixed wing can be used to support all or part of the weight of the aircraft in the higher speed range. The rotor, thus unloaded, will now be operating at a smaller value of C_T (Fig. 11) and higher values of μs, perhaps even beyond the limits of Fig. 11. Again, the aircraft is enabled to fly at a speed that uses the full power of the shaft-turbine.

Two-Speed Transmission – As mentioned in the previous discussion, tip stall is alleviated by increasing the rotor tip-speed. It is preferred, however, to have a lower tip-speed for good hovering and vertical climb performance. A shaft-turbine, however, at least the "solid" type, must be operated within a narrow speed range (compared with a piston engine). Thus it is necessary to select a gear ratio for the helicopter that is a compromise, because of the low tip-speed required for cruising flight. The use of a transmission that permits a change in gear ratio would eliminate the need for this compromise. The helicopter could then be designed to cruise at a speed high enough to use all the full power of the turbine and, at the same time, have good hovering performance.

This concept is shown in a somewhat different way in Fig. 12, which shows horsepower required plotted against forward speed for a series of rotor tip-speeds. If the only consideration were power required, it would be desirable to operate at as low a tip-speed as possible. The dash line, however, shows the limitation of blade stalling and clearly shows that the higher forward speeds can only be reached at high tip-speeds. The penalty in additional power for hovering at the high tip-speed, however, would be a significant decrease in vertical rate of climb. This penalty can be avoided by the use of a variable gear ratio transmission. The weight penalty and complexity of such a transmission must, of course, be considered in relation to the aerodynamic advantage obtained from its use.

Turbine Characteristics to Match Those of Helicopter

Every improvement that decreases the sfc of the shaft-turbine and closes the gap between reciprocating engines and gas turbines will help to establish more firmly the shaft-turbine in helicopter design. At the same time, it will increase the number of missions, the accomplishment of which is not always possible by helicopters equipped with present shaft-turbines.

Better Sfc at Part Load – Although the trend shows a general improvement in sfc of shaft-turbines, there remains the problem of high fuel consumption at part load. It is sometimes erroneously stated that helicopters require full take-off power for hovering and that, if hovering is a continuous operation, then sfc at take-off power is of paramount importance. This is only true under special operating conditions, such as when the helicopter is operated on a hot day (100 F), at a high altitude, or if the helicopter is marginally powered and has no reserve whatsoever for climbing. Modern helicopters are designed to have a vertical rate of climb under normal sea-level conditions of at least 800 fpm. This requirement in effect specifies excess installed hp to the extent of 20 to 25% of take-off power. Consequently, the power that is of most interest to the helicopter designer is the hovering power, which would be 75 to 80% of take-off power or possibly the maximum continuous power if the helicopter is designed for high-speed cruising, but certainly not the take-off rating. By proper helicop-

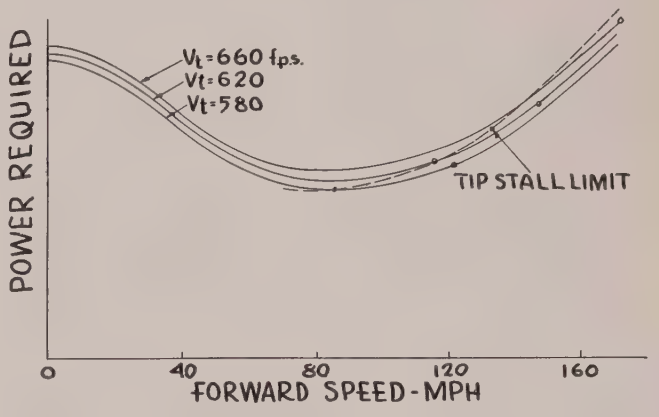

Fig. 12 – Helicopter performance limitations

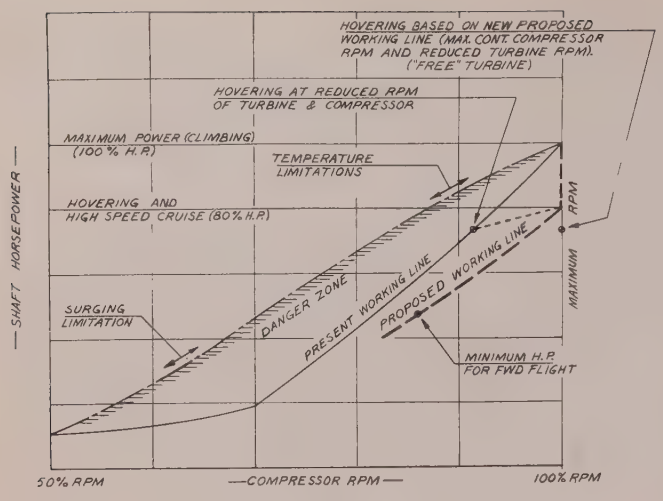
HOVERING BASED ON NEW PROPOSED WORKING LINE (MAX. CONT. COMPRESSOR RPM AND REDUCED TURBINE RPM). ("FREE" TURBINE)
HOVERING AT REDUCED RPM OF TURBINE & COMPRESSOR
MAXIMUM POWER (CLIMBING) (100% H.P.)
TEMPERATURE LIMITATIONS
HOVERING AND HIGH SPEED CRUISE (80% H.P.)
SURGING LIMITATION
DANGER ZONE
PRESENT WORKING LINE
PROPOSED WORKING LINE
MINIMUM H.P. FOR FWD FLIGHT
MAXIMUM RPM
— SHAFT HORSEPOWER —
50% RPM
—COMPRESSOR RPM—
100% RPM

Fig. 13 – Present and proposed typical shaft-turbine working lines

ter design, the power required for efficient cruising may be established in the same magnitude as that required for hovering (75 to 80% of take-off power). It then becomes obvious that a better sfc at part load is of special interest in the application of shaft-turbines to helicopters. The problem reduces itself to the question of whether it is possible to design the compressor and turbine to have their peak efficiencies at part-load condition so that the specific fuel consumption will be improved in this region, even at the possible expense of a decreased efficiency at take-off power.

Better Sfc for the Entire Helicopter Working Range – The second question that arises in the study of shaft-turbine characteristics is whether the shaft-turbine working line as a whole can be modified to fit better the aerodynamic requirements of the rotor, with the sfc changed accordingly. Fig. 13 shows a typical shaft-turbine power curve of hp plotted against rpm. The working line represents the optimum operational conditions. Operation at a higher rpm means a higher sfc. Operation at a lower rpm leads to overheating the turbine or causing the compressor to surge. The heavy dash line, on the other hand, shows the preferred operating rpm according to the helicopter rotor requirements. As more power is demanded at the higher flight speeds, the rotor must be operated at a high rpm to prevent stall.

Assuming full take-off power equals 100%, then hovering and V_{max} will be about 80% of take-off hp. For V_{max}, maximum rpm is desired to avoid stall of the rotor blades. Other points at lower power outputs are also determined by calculating the hp required for the particular forward speed at the corresponding rpm to avoid blade stall. For hovering, the rotor may operate at the same rpm as for cruising flight or, in order to improve the efficiency of the rotor, the rpm can be decreased. This will result in a decrease in hovering power until a point is reached on the shaft-turbine working line. For climbing, the power may follow either the working line or the constant-rpm line up to maximum hp. This shows the rpm limitation for a "solid" turbine. The helicopter designer would like to have the turbine working line moved over to approach as closely as possible the helicopter working line shown as the dash line on the graph. This will bring all the benefits of reduced sfc throughout the entire working range. There may be the penalty, however, of reducing the operational range of the gas turbine (especially at high power output) because the danger-zone line determined by high turbine temperature and compressor surge will follow the proposed working line. To obtain better efficiency in hovering, a reduction in rpm is required. With a "solid" turbine, this will be impossible because with the new working line any substantial reduction in rpm will bring the operation into the danger zone. Fortunately, with a "free" turbine the rpm of the compressor can remain at its optimum while the output turbine is reduced.

The question to be answered by the turbine designer is then: Can the shaft-turbine be designed closer to the helicopter working-line characteristics?

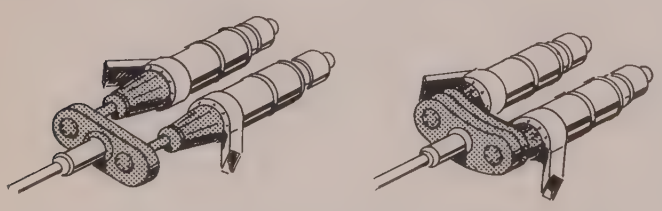

Fig. 14 – Left: individual engine reduction gearboxes and interconnecting helicopter gearbox; right: reduction and interconnecting gearbox in one unit

Preferred Design Features for Turbine Installation

"Free" Turbine – One of the best and most important improvements in the application of turbines to helicopters will be the "free" turbine arrangement. This has already been discussed earlier in the paper, and need not be repeated.

Reduction Gears Divorced from the Basic Engine – It was also stated earlier that it may be to the advantage of the helicopter designer to reduce torque and hence shaft weight by keeping the engine rpm high and reducing speed at the main gearboxes near the rotors. In multiengine design, the gearbox that interconnects the engines and provides a common shaft to drive the transmission may add unnecessarily to the total weight of the transmission. It

would be better to have the input to this combining gearbox operated at turbine speed so that the gearbox functions not only to combine the two engines but also to act as a reduction gearbox (see Fig. 14). These considerations lead to the conclusion that in general it is better for the helicopter designer to decide the reduction ratio and configuration of the speed-reducing gearbox.

It may also be pointed out that this concept makes a more flexible powerplant even for conventional airplanes. It permits the airplane designer to select a basic power unit, and use this unit in a number of different ways, each of which would require a different type of gearbox. He can use a single-rotation propeller, or two contrarotating propellers, or he can couple two or more engines to drive a common shaft, through overrunning clutches.

Provisions for Bleed Air – The compressor of the shaft-turbine should be provided with facilities for bleeding from one of the intermediate stages. This air may be used continuously to drive instruments and to supply low-pressure air to the penumatic system of the helicopter. It may also be used to cool the turbine wheel bearing, which cannot depend upon ram air as in the case of the airplane. Part-time utilization of this air might be for ventilation purposes. The hot section of the engine can be ventilated, and the engine and transmission oil cooled by an ejector system built around the exhaust pipe. However, the cold section of the engine should be prevented from accumulating an explosive mixture of air and fuel. In hovering flight when there is no ram pressure available to ventilate this compartment, air bled from the compressor may do the job. Bleed air may also be used to heat the inlet guide vanes for de-icing.

Arrangement of the Accessories – Accessories should be mounted on the top or bottom of the engine. (See Fig. 15.) In a conventional helicopter design, the engines are located either inside the fuselage or outside close to the fuselage skin. In both cases, lateral dimensions are critical and should not be unnecessarily extended. Moreover, in a multiengine installation, when the engines are placed side by side, accessories mounted on the sides of the engine will always be difficult of access. If, for any reason, some of the accessories must be mounted on the side, it is important to be able to mount them on either side. Otherwise, serious accessibility problems may arise, especially on multiengine designs.

Direction of Exhaust – The turbine design should be such that the exhaust can be easily ejected in any direction. In the usual solid turbine design, the exhaust pipe lies on the longitudinal axis of the engine and there is no particular problem with ejecting the exhaust in any required direction. The "free" turbine is likely to have a circumferential gas collector located between the output-turbine wheel and the output shaft or gearbox. In order that the turbine be adaptable to various helicopter

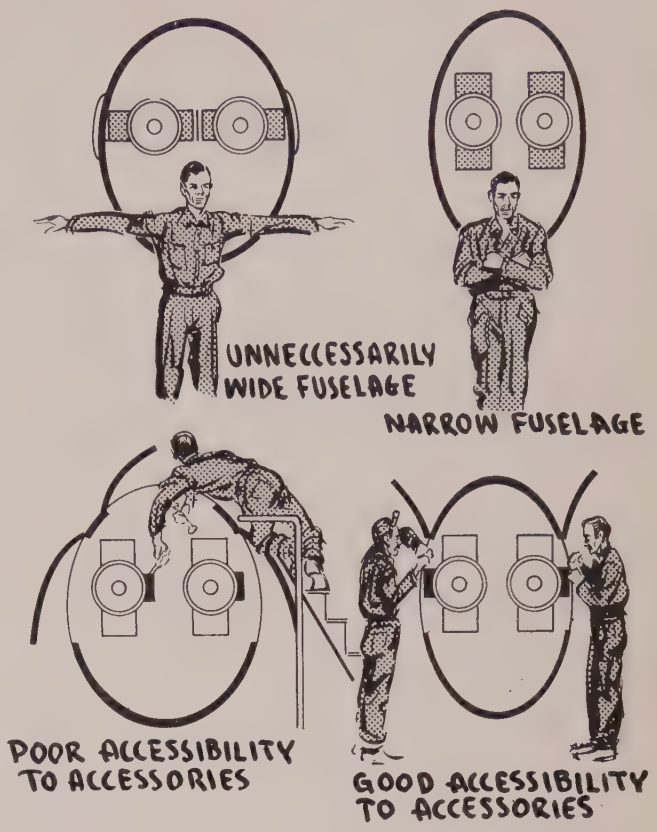

Fig. 15 – Effect of engine placement on width of fuselage and accessibility of accessories

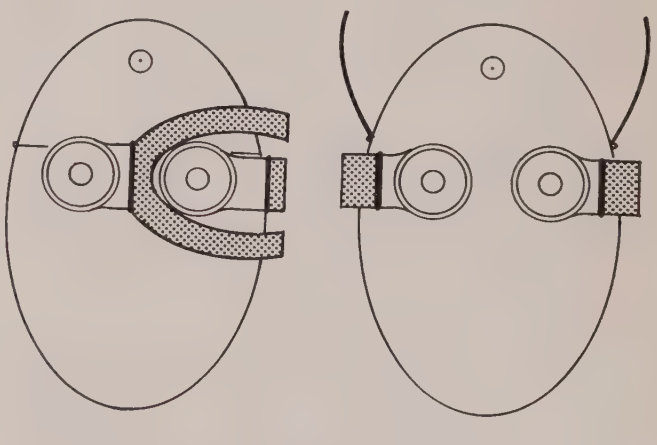

Fig. 16 – Left: Poor exhaust arrangement with long exhaust stacks protruding beyond fuselage contour; right: proper exhaust arrangement with short exhaust stacks providing enough clearance for engine removal

configurations and to various multiengine configurations, the collector ring should be designed to permit rotation of the exhaust duct to a favorable position, as shown in Fig. 16. Furthermore, the exhaust stack on the collector ring should be very short and provided with a flange and quick disconnecting clamping ring. This is important for easy engine removal.

Mounting – It is desirable to incorporate in the design of the shaft-turbine features that will enable the engine to be mounted in a vertical or steeply inclined position, in addition to the more conventional horizontal position. The modifications will probably be small if they are provided for in advance in the design of the turbine. They will be mostly associated with lubrication problems and with the strength of the front bearing. A flat face should be provided on the power-output end of the engine for mounting a clutch or gearbox. Provision should be made for attachment to the lightest type of engine mount. Since thrust loads are negligible in the absence of a propeller, it is necessary to design for torque and acceleration loads only. A maximum acceleration of 6g is considered adequate for helicopter engines.

Starting Methods – There is no special problem in starting shaft-turbines of low horsepower, since the battery normally provided for lighting, radio, and other electrical needs will be of sufficient amperage to operate the turbine starter. Turbines in the middle and high horsepower brackets, however, require so much starting power that the usual aircraft battery would have to be doubled or tripled, with a consequent severe weight penalty, unless starting is restricted to outside ground power sources. Such a restriction would render it impossible to start the engine under emergency conditions or at a place where outside power is not available.

With these larger shaft-turbines, it is customary to use an auxiliary power unit to supply starting power. For example, a small gas turbine compressor unit can supply compressed air to an air turbine starter motor. Unfortunately, such a unit is quite heavy and also takes up a substantial amount of space. This method of starting becomes more practical in a multiengine installation. In such a case, one power unit supplies compressed air for starting all of the engines. Only the air turbine motor need be provided for each engine along with the necessary ducting to the auxiliary gas turbine compressor unit.

Probably the best method for starting a shaft-turbine in a single-engine helicopter installation will be a combined cartridge and compressed-air starter. It is light, does not rely on a battery, and does not require a large space. Cartridges may be carried on board in sufficient quantities to take care of emergency starting. The cartridge starter and compressed-air starter can use the same type of starting motor. Starting air pressure of more than 100 psi may be supplied from ground carts. It would also be feasible to carry a small bottle of compressed air for emergency restarting.

Conclusion

It has been shown that the shaft-turbine is a superior helicopter powerplant for many applications. It will become even more superior with continuing improvements in specific weight and specific fuel consumption. There are, however, a number of helicopter characteristics as well as turbine characteristics that must be properly matched in order to produce an optimum design. In addition, there are a number of installation problems peculiar to helicopters, which should be considered in the initial design of shaft-turbines.

Bibliography

1. "Case for Turboprop Engines," by R. M. Hazen. Presented at SAE National Aeronautic Meeting, New York City, April 18, 1950.
2. "Development of Armstrong-Siddeley Mamba Engine," by W. H. Lindsey. *Journal of Royal Aeronautical Society,* Vol. 53, February, 1949, pp. 137-180.
3. "Principles of Jet Propulsion and Gas Turbines," by M. J. Zucrow. Pub. by Wiley, New York City, 1948.
4. "Gas Turbines for Aircraft," by F. W. Godsey, Jr., and L. A. Young. Pub. by McGraw-Hill, New York City, 1949.
5. SAE AIR No. 16, "Desirable Characteristics for Helicopter Engines," issued Feb. 1, 1947.

DISCUSSION

Shaft-Turbine Helicopter Has Low Noise Level

– Igor B. Bensen
Kaman Aircraft Corp.

WE are in a position to contribute to this paper in regard to the noise level of the shaft-turbine installation. Our experience with the shaft-turbine-powered helicopter indicates that the noise level is incomparably lower than that of a piston-engine installation.[a]

To be sure, the absence of vibration in the airframe greatly enhances the comfort of the passengers, but the absence of engine noise is almost startling to unprepared observers. The high-pitched tone of the compressor intake is rather low in volume, but is easily further attenuated by a simple muffler.

Exhaust stacks, on the other hand, produce very little noise since most of the energy of exhaust gases is extracted by the turbine wheels, leaving exit velocity quite low. This situation is typical for turboprop engines, and it has been the subject of many comments in technical literature here and abroad.

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[a] "Evaluation of Kaman Turborotor Helicopter," by J. O. Emmerson and S. R. Thompson. Presented before American Helicopter Society, Washington, May 15, 1952.

The Corvette Plastic Body

E. J. Premo, Chevrolet Motor Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1954.

IT is no secret in the automobile industry that interest in the sports car has gained tremendous momentum in this country during the past 10 years. Equally well recognized is the fact that, up to the present, this enthusiasm for a car that is "fun to drive" has been catered to almost exclusively by European importations, which, for the most part, are direct descendants of the race car and primarily adapted to driving conditions and economic factors abroad.

When Chevrolet decided to build the Corvette and to produce a new type of sports car on a volume basis there were many design factors to be considered. Essential sports car characteristics such as high performance and handling stability at all speed ranges had to be combined with the American public's conception of convenience, beauty, and a comfortable ride. Not least of the problems to be worked out was that of an ideal body design.

The Corvette (Fig. 1), first exhibited at the Waldorf Show in New York in January of 1953, was designed as a straightforward approach to a practical 2-passenger sports car.

With a height of 33 in. at the door, 48 in. over the top of the windshield, an overall length of 167 in., and an overall width of 72 in., it is not too large for a vehicle of 2-passenger capacity and still provides the passenger comfort that the American car owner is accustomed to, as well as adequate luggage capacity.

The Corvette body is the result of intensive research by Chevrolet on the functional application of reinforced plastics to the automotive field. The body shell, which weighs only 340 lb, is composed of glass fibers and polyester resins combined in a laminating process. The transparent plastic material has almost three times the thickness of steel and little more than half the weight of a comparable steel structure.

The material is rustproof and weathering tests equivalent to several years of exposure under the

THE story of the building of an automobile body from reinforced plastic is told here.

The body shell, which weighs only 340 lb, is composed of glass fibers and polyester resins combined in a laminating process.

Three methods of molding plastic parts are discussed here in detail:

1. **Hand-lay process.**
2. **Vacuum bag process.**
3. **Matched metal die process.**

The body parts are now being made by all three processes. Eventually, it is anticipated that all parts will be made by the matched metal die method.

The Author

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Fig. 1 – First Corvette built was exhibited in January, 1953

severest conditions have indicated no significant effect in the laminate strength or finish of the material. Another advantage of the plastic body is its silent riding qualities and relative freedom from drumming noise.

The eventual production model (Fig. 2) is virtually identical with the car that was displayed at the Waldorf Show.

Viewed from the front (Fig. 3) the car has a low, wide appearance. Two vertical bumpers attach to the chassis frame front cross-member to protect the radiator grille. Heavy-gage moldings are attached directly to the front fenders to protect them from casual damage. The headlamps are recessed into the front fenders and protected by stainless-steel wire-mesh screens. Cylindrical parking lamps, also used as direction signals, are located at each side of the grille.

The hood (Fig. 4) is hinged at the front to provide maximum safety and accessibility to the engine compartment. Parallel linkage hinges and hood lift springs provide the forward attachment, the function of the springs being to prevent rattle and give an initial lift to the front edge of the hood upon opening. The hood support is of the conventional star-channel type, which releases by lifting the hood, and a safety hole is provided in which a pin can be inserted to prevent accidental closing. Two locks of the smooth bar type, which do not require vertical adjustment, are provided at the rear of the hood.

The side view of the Corvette (Fig. 5) emphasizes length, with the high crown of the front fender flowing in an almost continuous line to the tail lamps. The single-cockpit styling creates a long rear deck line. The absence of an exterior side door handle adds to the streamlined appearance. Chrome-steel moldings extend entirely around the car.

The rear view (Fig. 6) reveals graceful, rounded body lines with styling emphasis on the car's wide, low-slung appearance. Jet-type tail lamps project from the tapering fins in the upper part of the fenders. Heavy-gage moldings, similar to those used in front, protect the rear fenders against damage. The rear bumper, spaced from the body by rubber blocks, is attached through the body

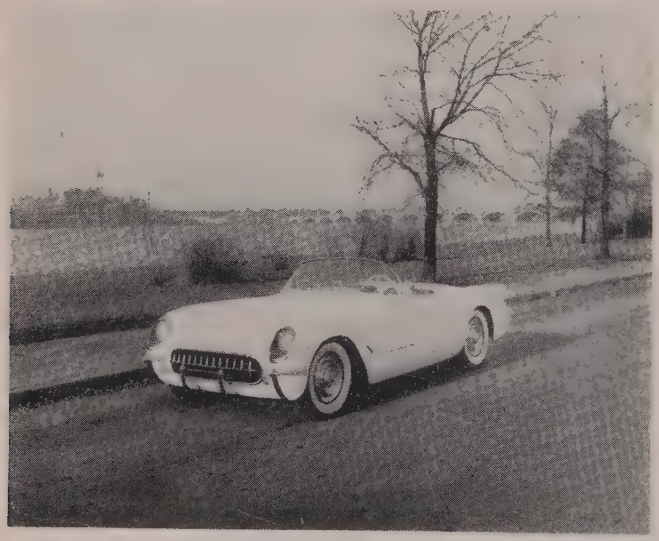

Fig. 2 – Production model of Corvette

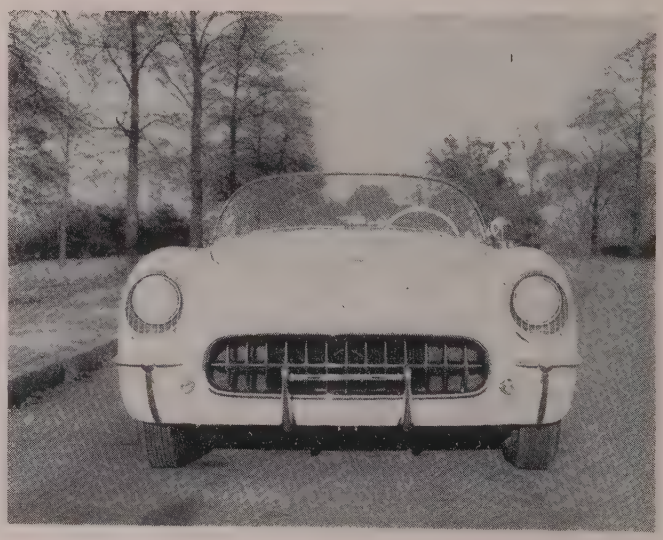

Fig. 3 – Front view shows low, wide appearance

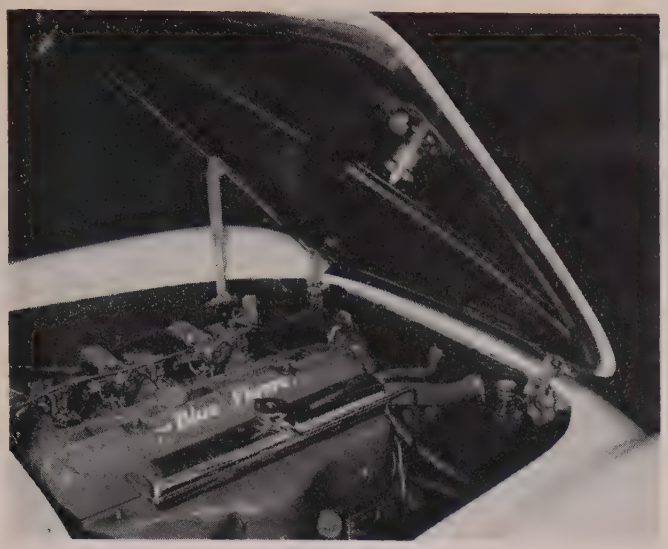

Fig. 4 – Hood is hinged at front to provide maximum safety and accessibility to engine compartment

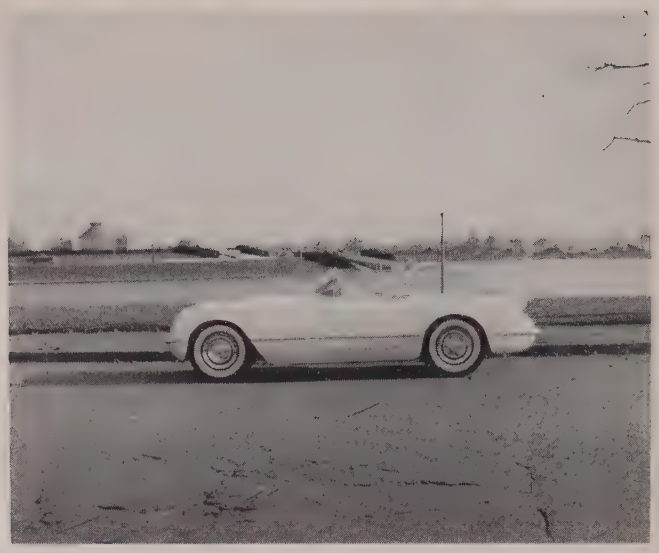

Fig. 5 – Side view emphasizes length with high crown of front fender flowing in almost continuous line to tail lamps

shell to brackets from the frame. The flexibility of the body material makes this a very practical design.

The dual exhaust ports are located at the ends of the bumper, the tail pipes extending through the body. The license plate is contained within a rectangular compartment in the rear deck lid and is covered by clear plastic.

The luggage compartment (Fig. 7) (approximately 10 cu ft in volume) has a tire well recessed into the floor, with a cover held in place by a bolt through the center of the lid. The luggage compartment lid, which is spring loaded to support it in open position, incorporates a wire-mesh-screen radio antenna.

The cockpit (Fig. 8) is fitted with two bucket seats separated by the driveshaft tunnel. The driver's seat is adjustable fore and aft while the passenger's seat is stationary. The transmission shift lever, connected directly to the transmission, is located on the tunnel, convenient to the driver.

The wide side doors (Fig. 9), which provide convenient entrance or exit to the cockpit, are fitted with storage compartments under the arm rest area. Ash trays are located at the forward end of the compartment covers.

Fig. 6 – Rear view reveals graceful, rounded body lines, with styling emphasis on wide, low-slung appearance of car

Fig. 7 – Luggage compartment has tire well recessed into floor, with cover lid held in place by bolt through center of lid

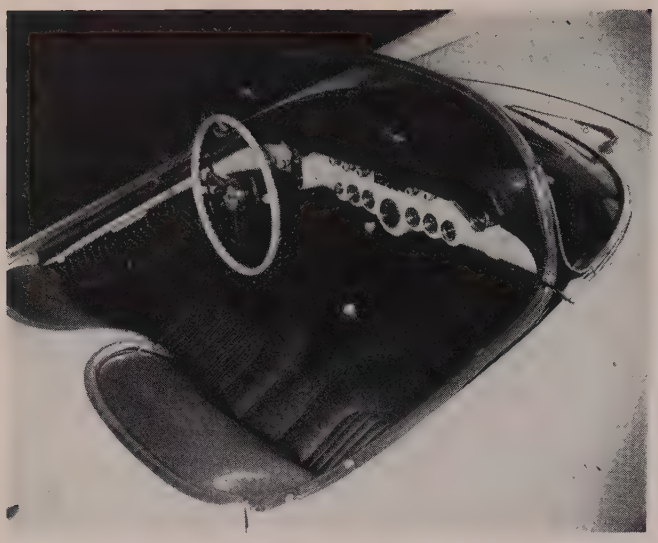

Fig. 8 – Cockpit is fitted with two bucket seats separated by driveshaft tunnel

The instrument panel (Fig. 10) includes the usual complement of instruments of the individual type, with the addition of a tachometer to indicate engine revolutions per minute.

The fabric top (Fig. 11) is stowed in a compartment at the rear of the cockpit. Space limitations required the development of a new design incorporating side bows, which fold toward the center of the car. The compartment cover hinges are coupled with the hinges for the luggage compartment lid to avoid interference between the cover and the lid if an attempt were made to open them both at the same time.

To raise the top (Fig. 12), it is first lifted out of the compartment and the cover closed. The folding links along the sides are opened and locked in position and the top is pulled forward for attachment to the windshield frame. Over-center hooks (Fig. 13) are provided at the front of the top to attach it to the windshield. The end of the hook is inserted in a hole in the side frame and the clamp is locked in position. The final operation (Fig. 14) is that of inserting the rear locks in the receivers provided in the stowage compartment cover.

Easily removable, clear plastic, chrome-framed side windows incorporating ventipanes are provided to complete the enclosure. To remove these windows (Fig. 15), a knob inside the door is operated to release the door bracket and the window is tilted toward the hinged side of the door to remove the front bracket from its receptacle. A side window storage container is provided in the luggage compartment.

We had about six months of development work behind us at the time the first Corvette model was presented at the Waldorf Show. The body on the show model was made of reinforced plastic purely as an expedient to get the job built quickly.

This reinforced plastic was composed of resin-saturated glass fiber cloth which was laid-up in plaster forms obtained directly from the clay model. The material thickness was closer to 0.2 in. than to the 0.1 in. eventually adopted because it was comparable to a steel body in strength. The combination of glass cloth and mat had a rigid resin of 0.5% elongation at the rupture point.

This resin was not sufficiently flexible to utilize

Fig. 9 – Wide side doors are fitted with storage compartments under arm rest area

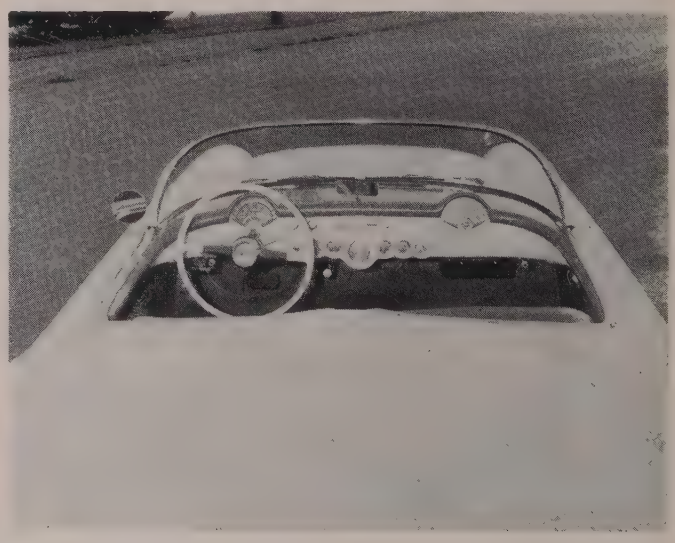

Fig. 10 – Instrument panel includes usual instruments plus tachometer to indicate engine rpm

Fig. 11 – Fabric top is stowed in compartment at rear of cockpit

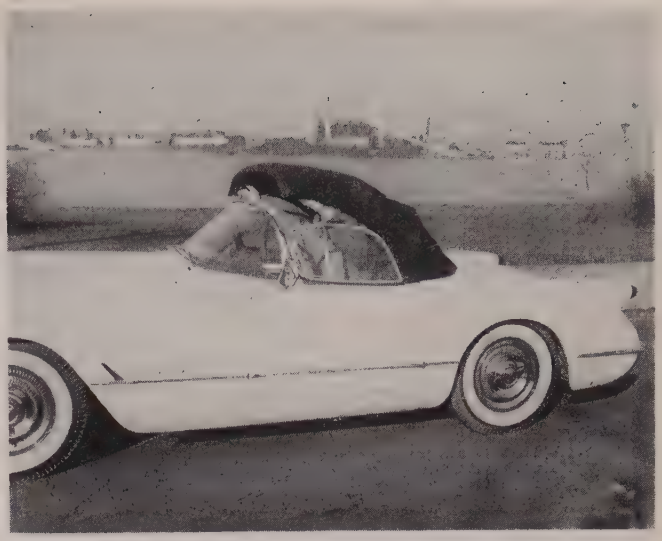

Fig. 12 – Top being raised

the strength of the fiber glass, which had an elongation of approximately 3% at rupture, and it was necessary to refinish the model several times during the various displays across the country because of crazing of the resin, which showed through the paint.

Although we were going ahead with the building of an experimental plastic body (using 5.5% elongation resin) in order to get a car rolling for chassis development work, at the time of the Waldorf Show we were actually concentrating body-design-wise on a steel body utilizing Kirksite tooling for the projected production of 10,000 units during the 1954 model year. It was some time later that we decided to produce this quantity in reinforced plastic.

The public acceptance of the Corvette had been immediate and overwhelming and there was no denying the demand that existed for an American-built sports car. People just would not wait until 1954 – they wanted delivery right now, and so our management decided to build 300 Corvettes with plastic bodies during 1953, starting in June. This meant that we had five months to complete a suitable plastic body design and tool it for production, an undertaking that was not unreasonable using the hand-lay or bag method.

About this time, some doubt was expressed that

Fig. 13 – Over-center hooks are provided at front of top to attach it to windshield

Fig. 14 – Inserting rear locks in receivers provided in stowage compartment cover

Fig. 15 – Chrome-framed side windows being installed

we should build the 1954 model of steel. People seemed to be captivated by the idea of the fiber glass plastic body. Furthermore, information being given to us by the reinforced plastic industry seemed to indicate the practicability of fabricating plastic body parts for automobiles on a large scale.

To survey this situation and to determine whether or not the capacity to fill our requirements existed in the industry, we set out on a tour of reinforced plastic manufacturing plants.

Field Survey

Our first stop was at a fabricating source, which was using the matched metal die method in producing machine housings, bakers' trays, flat sheet stock, and other miscellaneous parts.

Fig. 16 – Molds used in matched metal die method

This manufacturer stated the matched metal die process would produce parts suitable for automobile bodies at the rate of 100 pieces per machine per 8-hr day. Although his firm did not have the capacity to meet our needs at the present location, he said that machines and tools could be manufactured in four to eight months. The tooling time would, of course, depend upon sizes of the pieces involved.

This company (Fig. 16) uses molds made from iron castings or high carbon plate steel, depending on the depth of the part to be made. The male and female halves are accurately machined to provide a space between them equal to the desired thickness of the finished part and the surfaces are highly polished. A flame-hardened pinch-off is provided at the edges of the part. Fitted to within one to two thousandths of an inch, it cuts off the excess of fiber glass preform and seals the edges of the mold to provide hydraulic pressure in the molding operation. To cure the part, steam under pressure at a temperature of approximately 240 F is circulated through the dies.

The molds are fastened into a column press capable of exerting a pressure of from 100 to 150 psi. Prior to the molding operation, the molds are coated with a parting agent. After the dies are worn in, an application every 40 or 50 pieces is all that is necessary.

Perforated metal preform screens, the shape of the inside of the finished part, are used for making the fiber glass preforms. One of these screens is mounted on a rotating table of a machine incorporating a strong exhaust blower, which sucks air through the holes.

Fiber glass roving, cut in short lengths and deposited on the form by means of a blower, is held in place by this suction. The form is rotated during the process to assist the spreading of fibers; periodically, a small amount of resin is sprayed on to hold the resulting preform together, as shown in Fig. 17.

The amount of fiber glass deposited is controlled by an electric timer and the uniformity of the preform is the result of the operator's performance. Several screens are used and as the preform is finished (Fig. 18) it is placed together with the screen in an oven at 350 F for a period of approximately 3 min to cure the binder.

The polyester resin is mixed with an inert filler such as calcium carbonate to reduce shrinkage or crazing, impart a smoother finish, and to reduce the overall cost of the mixture. Activating and accelerating agents are then added to facilitate the curing of the resin. The ingredients are thoroughly mixed in relatively small quantities just prior to using, as shown in Fig. 19.

The preform is placed in the mold (Fig. 20) and a measured amount of resin mix is spread over it.

Fig. 17 – Fiber glass preforms being made

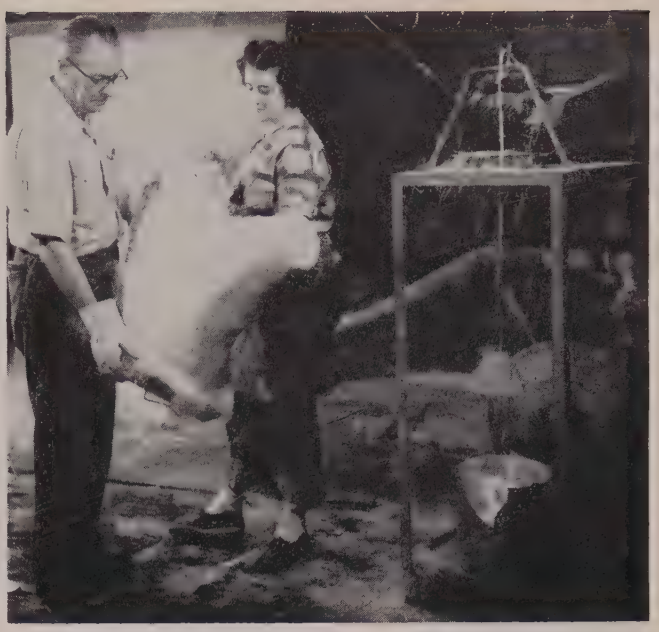

Fig. 18 – Fiber glass preform and screen ready to be placed in oven

Fig. 19 – Resin mix being produced

Fig. 20 – Preform being placed in mold

The mold is then closed for an electrically timed curing period of approximately three minutes, after which it automatically opens and the part is removed, as shown in Fig. 21. Excess roving and flash around the edges, which does not completely pinch off, are easily removed with an electric sander, as shown in Fig. 22.

We were quite impressed by this process and the possibility of production of over 20,000 parts per year from one set of tools, operating an 8-hr day.

Our second stop was at a company that was using the pressure bag method in producing plastic boats.

This company also believed that fiber glass reinforced plastics would make a suitable material for automobile bodies, and pointed out that any material having the ability to withstand the severe buffeting that motor boats were subjected to should certainly meet our requirements. He cautioned us, however, against overoptimism and offered his assistance in overcoming any obstacles, in our common interest.

We found this same spirit of cooperation throughout the reinforced plastic industry. It was the general feeling that the success of our undertaking would accelerate their development by at

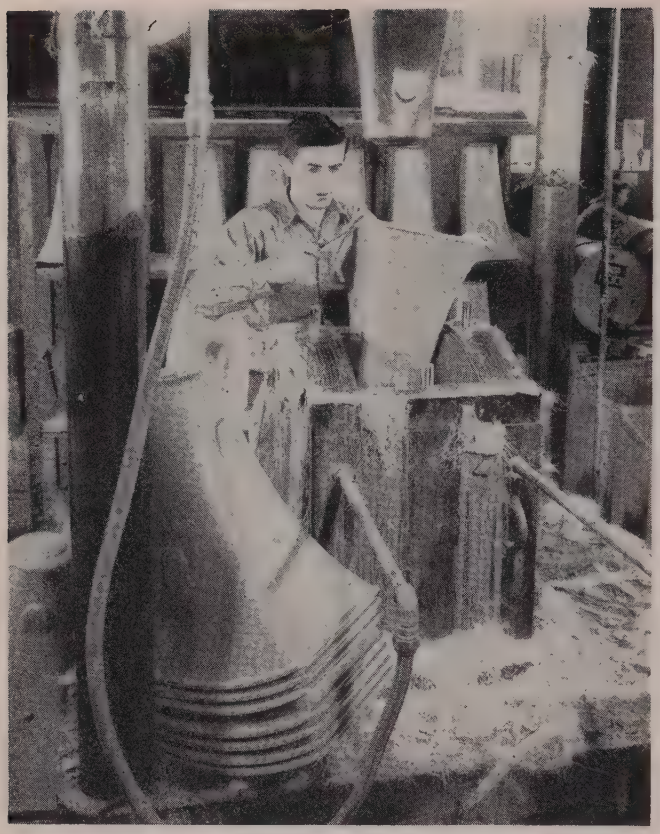

Fig. 21 – Part being removed from mold

least five years and, on the other hand, the failure of our undertaking would set them back for at least as much.

Fiber glass mat is usually used at this plant, although fiber glass cloth or preforms could also be used with the pressure bag process. Several thicknesses of the mat are spread on a table (Fig. 23) and parts to fit the molds are cut by power scissors. The cut mat is arranged in sets, each set having enough parts to make up a boat.

The molds in use here are constructed of upper and lower halves, which clamp together into a unit capable of withstanding the 50-psi bag pressure. Since it is desired to obtain a smooth outer surface on boat parts, the female half of a boat mold is made of polished aluminum. The rubber bag, which provides the pressure, is attached to the male half. (See Fig. 24.)

After a coating of parting agent has been applied to the aluminum surface of the mold, a surface of pigmented flexible polyester resin, approximately ten to twenty thousandths thick, is sprayed on, as shown in Fig. 25. While the fiber glass mat is being laid in place, sufficient polyester resin to saturate the mat is applied by a spray gun (Fig. 26), which mixes the activator with the resin at the spray nozzle. The number of layers of mat applied can vary throughout the part being made and reinforcements of plywood or other suitable material may be used with the pressure bag method, as shown in Fig. 27.

When the laying-up has been completed, a sheet of polyvinyl alcohol, known as pva, is spread over the surface (Fig. 28) to prevent adhesion of the layup to the bag. This material does not bond to the polyester resin and is very easily removed after the part has been cured. In fact, the operators use this material as aprons to protect their clothing. The bag is applied carefully (Fig. 29) to ex-

Fig. 22 – Excess roving and flash being removed from edges with electric sander

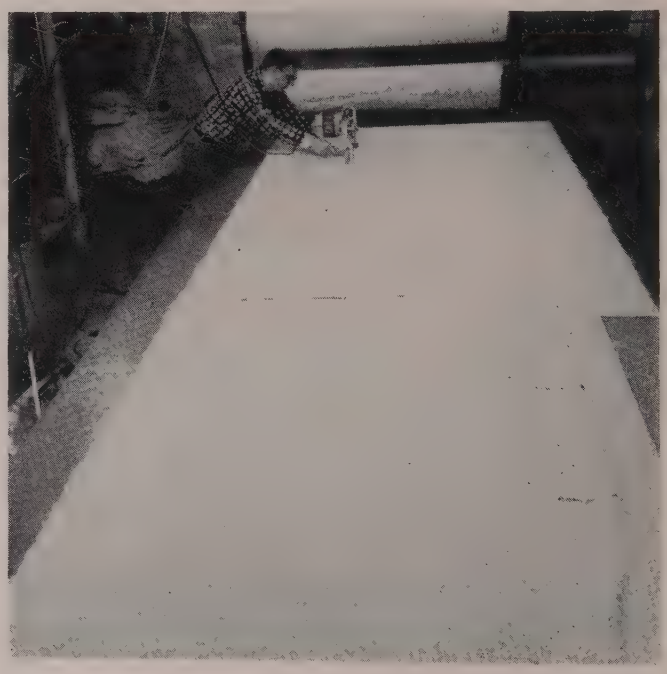

Fig. 23 – Cutting glass fiber mat into shape for molding operation

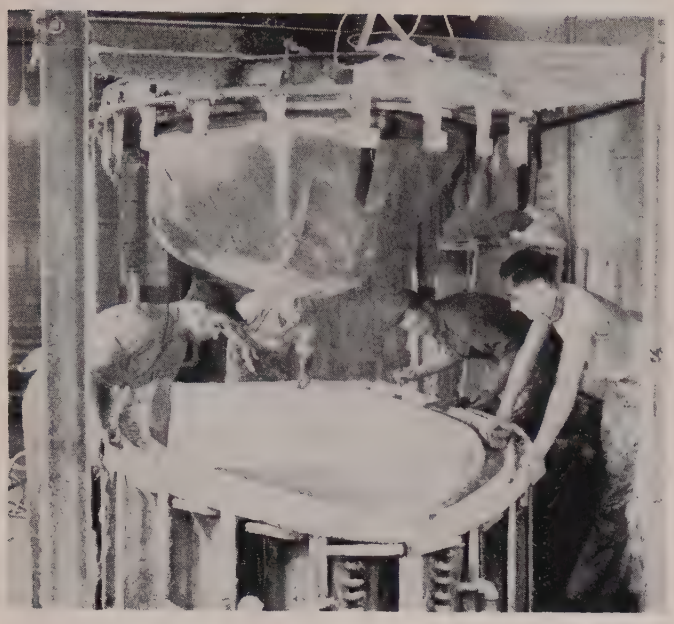

Fig. 24 – Upper half of boat mold with rubber bag attached and lower half, which is made of polished aluminum

Fig. 25 – Pigmented flexible polyester resin being sprayed on aluminum mold

Fig. 26 – Polyester resin being sprayed on fiber glass mat

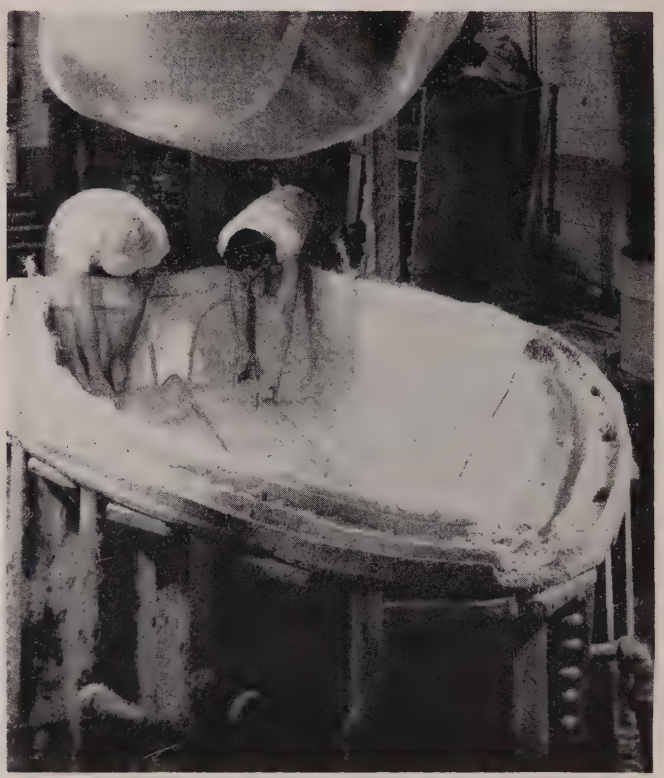

Fig. 27 – Plywood reinforcements being added to mold layup

Fig. 28 – Spreading polyvinyl alcohol sheet over surface of layup

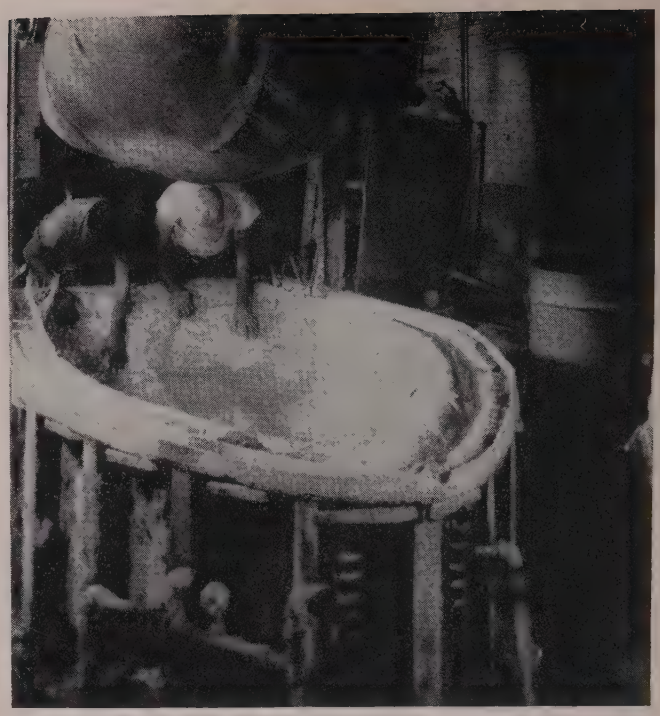

Fig. 29 – Polyvinyl alcohol bag being applied carefully to mold to prevent trapping air

clude as much air as possible and the top edge is trimmed around the top of the mold to conduct any excess of resin into the trough provided.

The mold is closed, clamped securely (Fig. 30), and the bag is inflated by air to 50-psi pressure. Hot water is circulated through the lower part of the mold to complete the curing. After a heat cycle of approximately one hour, the hull is removed from the mold and the flash is trimmed by means of a cut-off grinding wheel. The edges and surface are sanded (Fig. 31) to remove any defects and the part is completed.

Decked boats (Fig. 32) are made of two pieces bonded together. By using contrasting color pigments in the resin, very striking results are obtained. One of the late models utilizes an external flanged joint as a spray rail. This very practical type of joint is also used completely around the Corvette and is covered by a trim molding. It is interesting to note the very intricate shape (Fig.

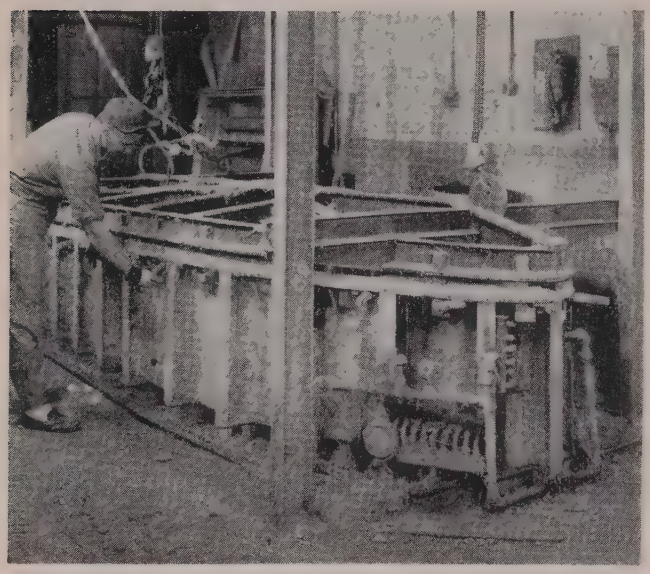

Fig. 30 – Clamping molds together for curing operation

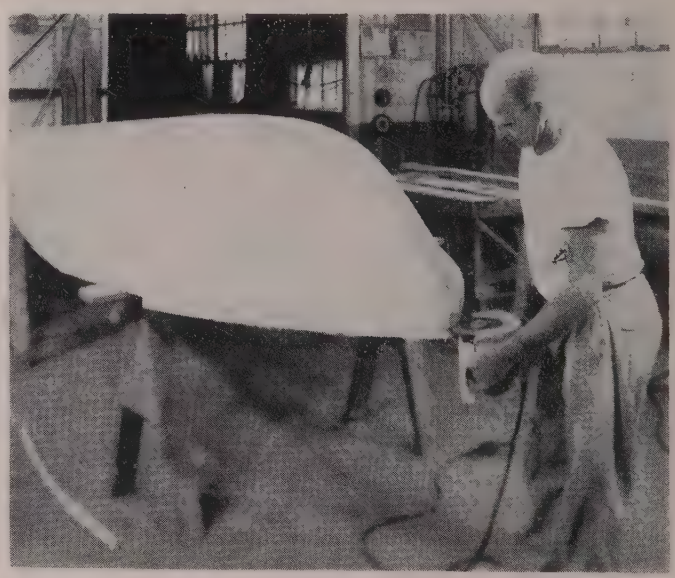

Fig. 31 – Flash being trimmed from hull with cut-off grinding wheel

Fig. 32 – Completed decked boats

Fig. 33 – Close-up showing intricate shape of molded boat

33) that it is possible to mold in one piece using the pressure bag method.

This company was also manufacturing parts by the matched metal die method (Fig. 34) when they were of suitable size and volume was sufficient to warrant the higher tooling costs involved.

At the third stop in our field survey, we visited a company that used the vacuum bag method of molding reinforced plastic parts.

We were advised by this company that, since starting in the reinforced plastic industry in 1946, they had manufactured a variety of items ranging in size from a small, 1.5-in. electrical terminal block to 8-ft roof top ventilators. These items include reinforced plastic horses and rabbits used in amusement parks, floats for sulfuric acid tanks, bath tubs, caskets, and protective baseball caps.

An experimental mold made of layers of fiber glass cloth impregnated with polyester resin, which had been formed directly over a production metal part, was used to demonstrate the vacuum bag process.

In this process the surface of the mold is coated with wax and then sprayed (Fig. 35) with a water

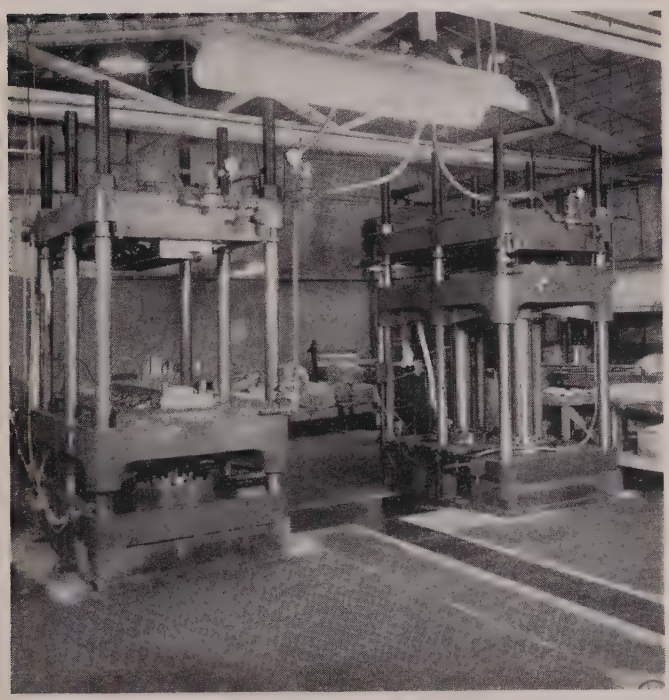

Fig. 34 – Presses used in matched metal die method

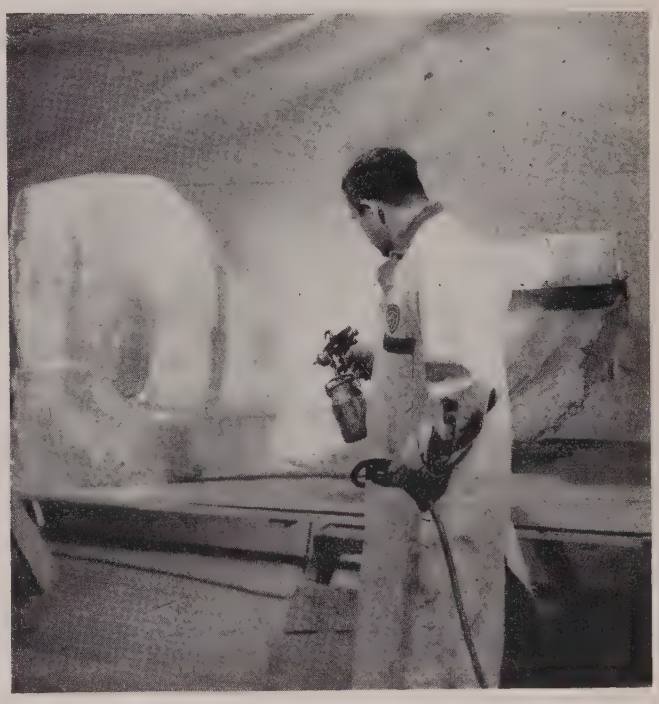

Fig. 35 – Experimental mold being sprayed with polyvinyl alcohol for a parting agent

Fig. 36 – Formulated resin being sprayed on mold for surface coat

Fig. 37 – Activated polyester resin being spread over mat with wooden paddle

Fig. 38 – Resin-impregnated mat being laid in mold

Fig. 39 – Felt bleeder strip is taped to mold to provide air escape

emulsion of pva as a parting agent. This material has a blue dye added in order to distinguish the areas that have been sprayed. The color is also useful to determine when the parting agent has been completely removed from the finished part, since it will prevent good paint adhesion.

Formulated resin to which white pigment has been added is sprayed (Fig. 36) on the mold to a thickness of approximately 0.010 in., this, of course, being the gel coat, which provides a smooth surface on the finished part. After curing for approximately 15 min, the coating is tacky and ready for the layup of the fiber glass mat.

Three layers of precut 2-oz per sq ft glass mat are spread on a table over a film of pva. Activated polyester resin (Fig. 37) is then spread over the mat using a wooden paddle to provide even dispersion of the resin and to remove air bubbles. The resin-impregnated mat is now laid up in the mold, as shown in Fig. 38. The material is smoothed out by hand and the joints are carefully lapped. A strip of felt material is taped around the edges of the mold at the trim line to provide an air escape, as shown in Fig. 39. Absorbent material is located at strategic points to soak up the excess resin squeezed out of the mat during the vacuum operation.

A sheet of pva, known as the bag, is spread over the layup and strips of wood to which rubber has been cemented are placed around the edges of the mold and clamped in place to provide a seal. Although numerous clamps are used in this limited production process, when a quantity of parts is to be made, clamping members (Fig. 40), which are more readily applied, are provided.

Holes are punctured in the bag at several places in the vicinities of the felt air bleeder strip and the absorbent material locations, and vacuum lines are connected to the bag. The air between the bag and the material layup is removed by means of a vacuum pump, thus applying atmospheric pressure to the part.

Air bubbles are worked toward the edge of the mold by hand, using wooden paddles, while the air is being exhausted. (See Fig. 41.) The length of time required to cure the part depends on the amounts of activator and accelerator in the resin mix and whether or not heat is applied.

It was quite apparent that this experimental process involved low tooling costs and enabled the production of a small quantity of parts in a very short time.

To complete the survey of manufacturing methods, we then visited a boat manufacturer to investigate hand layup methods.

The owner informed us that his boat building experience dated back to 1931 when he first started manufacturing boats from wood, and that he started using reinforced plastic materials in 1945. He expressed the opinion that hand layup would be a simple and fast way to manufacture a pilot model car or even produce cars on a limited scale, since the tool cost would be low.

Fig. 40 – Strips of wood being clamped in place to seal bag at mold edges

Apparently, size presents no problem in his operations as he manufactures boats ranging in size from a small 8-ft dinghy to a 28-ft sloop.

We were invited by him to observe a typical hand layup and a 12-ft-boat female mold was selected for the demonstration. (See Fig. 42.) This mold was made of fiber glass mat and polyester resin with a plywood box framework to hold it in shape.

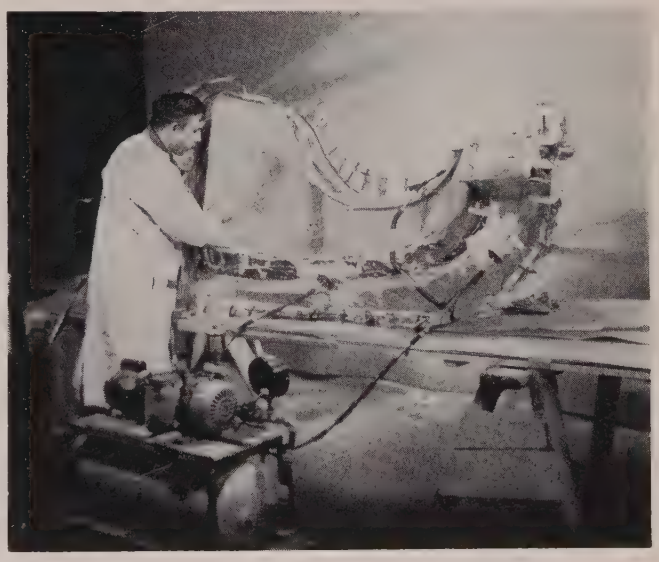

Fig. 41 – Excess resin and air bubbles being worked toward mold edges

Fig. 42 – Fiber glass mat being applied to female mold for 12-ft boat

In the hand-lay process as employed by this company, the mold surfaces are waxed, sprayed with a water emulsion of pva and a gel coat is added to provide the finish surface.

The curing of the gel coat to a tacky condition is sometimes accelerated by electric strip heaters, placed inside the mold. Fiber glass mat that has been cut to shape is applied and smoothed out by hand. The mat is saturated with activated polyester resin (Fig. 43), using an ordinary paint brush to spread it and work out any trapped air bubbles. Alternate layers of resin and glass mat are applied until three layers have been completed. The top of the mold is then covered by a plywood panel to which electric strip heaters are attached, as shown in Fig. 43. The temperature of these heaters is thermostatically controlled to 220 F.

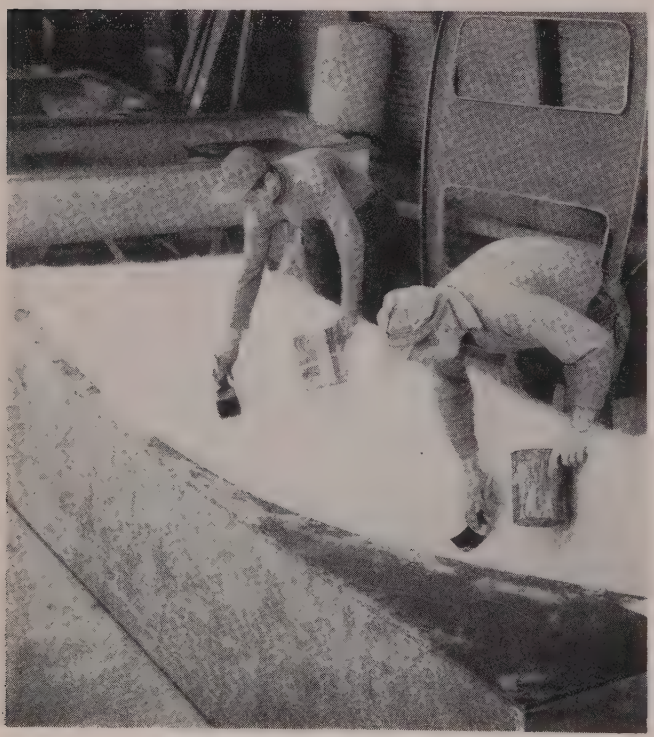

Fig. 43 – Mat being saturated with activated polyester resin

To retain the heat from the strip heaters inside the mold, a canvas cover is spread over the entire unit, as shown in Fig. 45. Here, again, the curing time depends on the accelerator in the resin mix and the amount of heat used. After the part is cured, it is removed from the mold, washed with a solvent to remove the parting agent, and trimmed.

The deck for this boat is fabricated in the same manner as the hull, using a resin mixture that has been pigmented to a contrasting color. The deck is bonded to the hull along the flange and the lap joint is covered by an aluminum rub rail attached with sheet-metal screws. The seats are subassembled and bonded to brackets which were formed in the hull at the time of layup.

This method was certainly simple and was very similar to the process used for the experimental cars exhibited at the Waldorf Show, with the exception of the mold, which, in this case, was made of reinforced plastic instead of plaster to extend the life of the mold.

Summary – In summarizing the results of our investigation, we first reviewed the materials used. We found fiber glass, in the following basic forms, suitable for our purpose:

1. Roving, an accumulation of strands into a loose, rope-like form, in spools.
2. Fiber glass mat, a nondirectional accumulation of strands into a mat in rolls of various widths.
3. Fiber glass cloth, consisting of glass fibers woven into a fabric.

The same type of polyester resin is used with all processes. It is available from many different sources and is a light-colored liquid resembling corn syrup. It can be stored at room temperature for approximately four months and still retain its original properties.

It can be extended up to 25% with bulk fillers such as calcium carbonate, talc, clay, magnesium silicate, and many other materials, to reduce cost, and will actually gain by this addition in some characteristics, such as reduction of shrinkage and crazing, improved surface finish, and increased strength and rigidity.

Upon addition of an organic peroxide, known as the catalyst, and the application of heat, the resin is cured or hardened. Certain other chemicals, known as accelerators or promoters, are also used to reduce the curing time.

Body Design

The Corvette body panels were designed to meet

the requirements of the matched metal die process. We used the same design for the hand-lay and vacuum bag tooling, so that a gradual changeover could be effected as each matched metal die was completed.

The general body panel breakup is shown in Figs. 46 and 47. The largest single piece is the underbody, which includes the entire floor from the front to the rear, the toe pan surface, and the lower portion of the dash panel. The lower part of the rear quarters and the rear panel are also included up to the widest point of the body, which provides a natural parting line. The shape of the underbody necessary to clear the chassis provides a great deal of the stiffness required, and few reinforcements are needed.

The arrows, laid out on the underbody, point to the locations of the body attachments to the frame, of which there are 11.

No. 1 outer l-h and r-h body bolts are located at the junction of the toe panel and the floor panel adjacent to the hinge pillars. No. 1 inner l-h and r-h body bolts are also located at the junction of the toe pan and floor pan, but at each side of the tunnel area.

No. 2 outer l-h and r-h body bolts are located just forward of the bulkhead behind the seats, close to the body sills. No. 2 inner l-h and r-h body bolts are also located just forward of the bulkhead, but farther inboard. Of these eight bolts, the four outer ones attach to outboard brackets welded to the frame side rails, and the remaining four inner bolts attach to the frame X-member near the ends, where it attaches to the frame.

Fig. 44 – Top of mold being covered with plywood panel having heating element

The chassis frame provides the major portion of the beam and torsional strength and rigidity through the center section of the car. The rear portion of the body is also attached to the frame at the outer ends of the rear cross-member as indicated by the arrows.

The eleventh point of attachment is at the center of the radiator support.

The outer rocker panel includes the lower por-

Fig. 45 – Canvas cover being placed over entire unit to act as seal

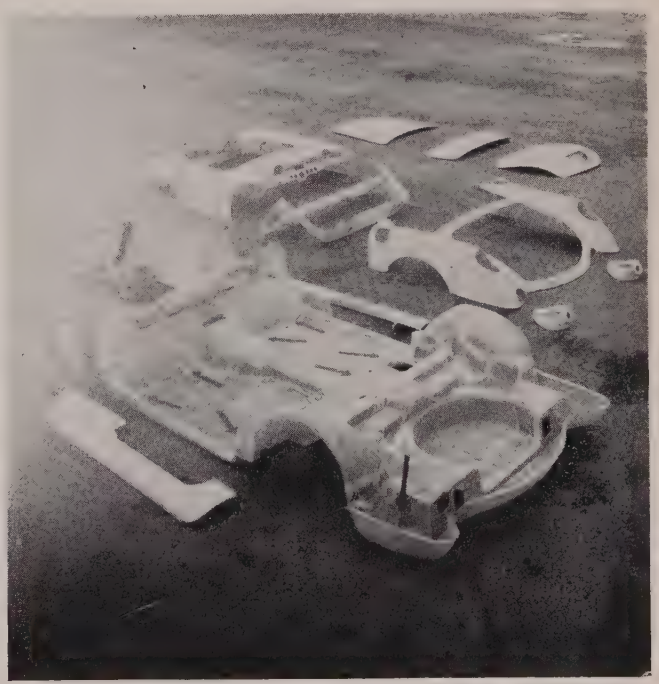

Fig. 46 – General view of actual body parts layout

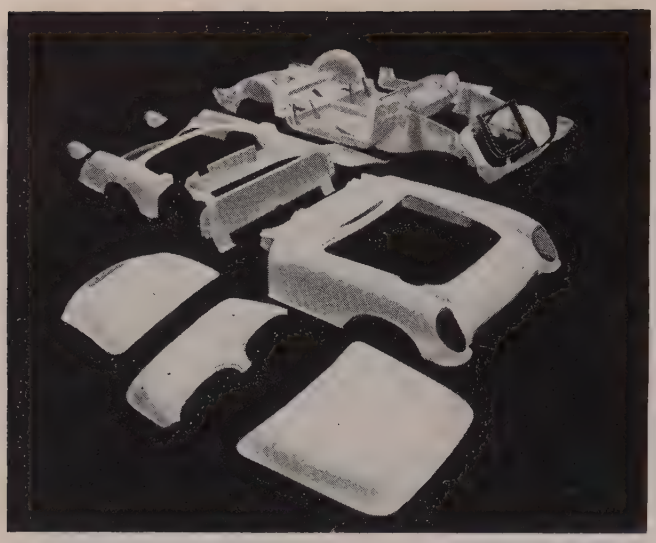

Fig. 47 – Close-up view of body parts layout

tion of the side of the body between the front and rear wheel openings and up to the widest point of the body.

The dash panel upper extends from the top of the dash portion of the underbody to the outer skin.

The left-hand and right-hand fender skirts extend from the dash to the radiator support bulkhead, which is made up of left-hand and right-hand front fender skirts and upper and lower filler panels. The steel radiator support forms a framework around the opening in this bulkhead.

The front upper panel extends from the horizontal center of the radiator grille at the front to the passenger cockpit, including the top of the instrument panel. It also includes the sides of the body from the door opening forward, usually considered as front fenders, downward to the widest point of the body.

The rear upper panel extends from the passenger cockpit to the rearmost point on the body and includes the quarter panels down to the widest part. This part also includes a portion of the lock pillars.

The openings for the hood and the luggage compartment cover are contained entirely within the front and rear upper panels. This makes it much easier to maintain the size and shape of the openings than when several panels are used.

The bulkhead includes the top stowage compartment area and also the remaining portion of the lock pillars. The gasoline tank is located behind this bulkhead and a cover is provided for access to it.

The front lower panel includes the lower portions of the front fenders and completes the radiator grille opening. The hood assembly is made of an outer panel and an inner skeleton reinforcement, which forms a box section entirely around the perimeter and across the center. This same type of construction is used for the top stowage compartment and luggage compartment covers.

The side doors are of 2-piece construction, having an outer and inner panel.

The instrument panel lower includes provision for mounting the individual-type instruments used and is attached to the rear edge of the front upper panel and to the hinge pillars.

This total of 24 major pieces makes up the greatest part of the body weight. The total weight of the Corvette reinforced plastic body parts is 340 lb, the basic materials making up this total being 136 lb of fiber glass, 153 lb of polyester resin, and 51 lb of inert filler.

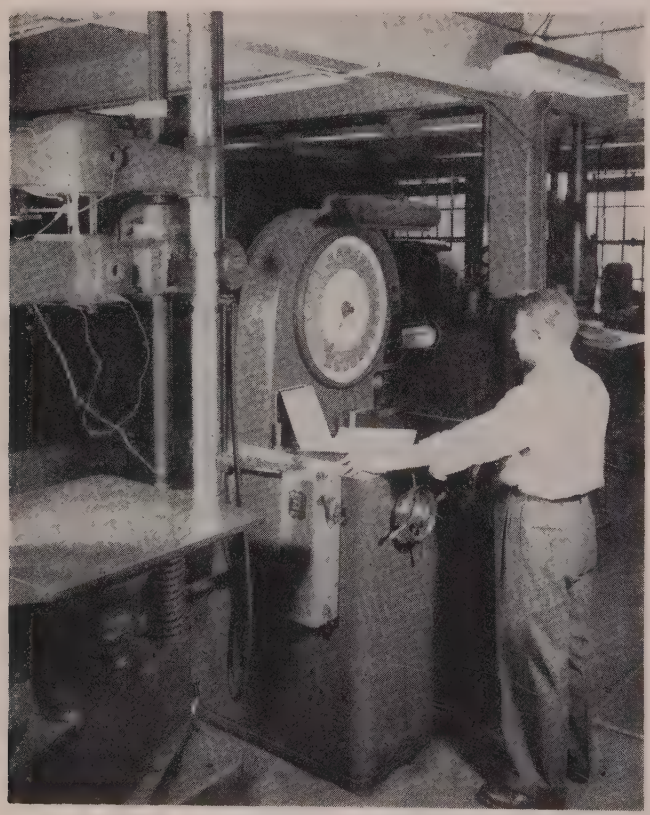

Fig. 48 – General-purpose tensile test machine

Material Specifications

Specifications were being developed to control the quality of the parts while the design work was in progress. Methods of evaluating the material in the laboratory were also required.

We approached the problem of preparing a specification for the reinforced plastic material with the object of obtaining physical properties equivalent to the panels being used for the steel body.

Samples made by the matched metal die process were obtained that were composed, by weight, of 40% fiber glass roving and 60% resin mixture. Twenty-five per cent of the resin mixture, or 15% of the total composition, was filler. From tests of

these samples it appeared that 15,000-psi minimum tensile strength was realistic. Using these figures, we calculated that reinforced polyester resin 0.1 in. thick would be approximately equivalent in tensile strength to 0.036-in. thick steel.

The general-purpose tensile test machine shown in Fig. 48 appears somewhat large for the small sample being tested but it is extremely sensitive and incorporates a recorder at the right, which plots a stress-strain curve automatically.

We specify a minimum tensile modulus of elasticity of one million psi, which is the highest obtainable at present. Even considering the thickness difference, this value is only 1/10 of that required to be equivalent to steel, but this lack of rigidity is offset, to a large degree, by the superior damping qualities of the reinforced plastic. An ordinary beam strength test setup (Fig. 49) is used to check flexural strength.

A minimum of 20,000 psi has been found practical for fiber glass reinforced polyester resin 0.1 in. thick. We require this minimum at room temperature and are at present determining a minimum value at 180 F.

The flexural wet strength of reinforced plastic is measured after boiling a test sample 2 hr in water and allowing it to cool in the liquid to room temperature. We specify a minimum of 18,000 psi, or 70% of the dry flexural strength at room temperature, whichever is the greater. For example, if the room temperature strength is 28,000 psi, then the minimum wet strength acceptable is 70% of that figure, or 19,600 psi. In other words, the material must not lose more than 30% of its strength. It is not anticipated that this figure will ever be exceeded in normal usage.

Although the flexural modulus of elasticity can also be measured directly by a beam test at room temperature, this method is not practical at elevated temperatures. Accordingly, we use a device that measures the torsional modulus of elasticity directly and then calculate the flexural modulus. The results are comparable. The equipment consists of a dead-weight loading arrangement applied through a freely turning, graduated pulley, which is connected to one end of a test sample, the other end of the sample being rigidly clamped. It will be noted in Fig. 50 that at room temperature three weights have caused the sample to twist only slightly.

This method is used because of the advantage of being readily adapted to elevated-temperature measurements, since the sample clamped for test can be entirely immersed in a suitable heating medium such as ethylene glycol. Fig. 51 shows the extreme amount of twist in the test sample resulting from a single weight at elevated temperature.

For the 0.1 in. thickness of reinforced plastic to be equal in flexure to the 0.036-in. thick steel, the flexural modulus of elasticity would have to be 1,500,000 psi. This figure is obtainable at room temperature, provided we do not use too flexible a resin.

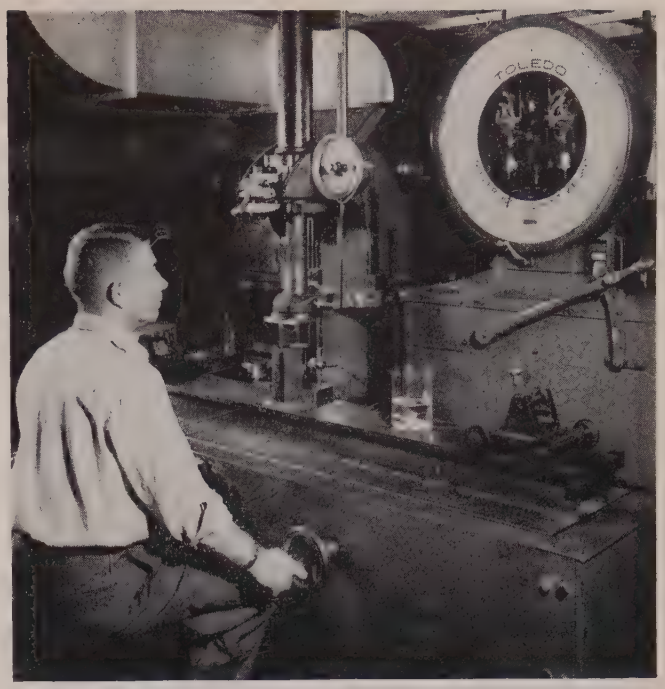

Fig. 49 – Beam strength test setup used to check flexural strength

To control the rigidity of the plastic panels at elevated temperatures, a test point of 180 F was

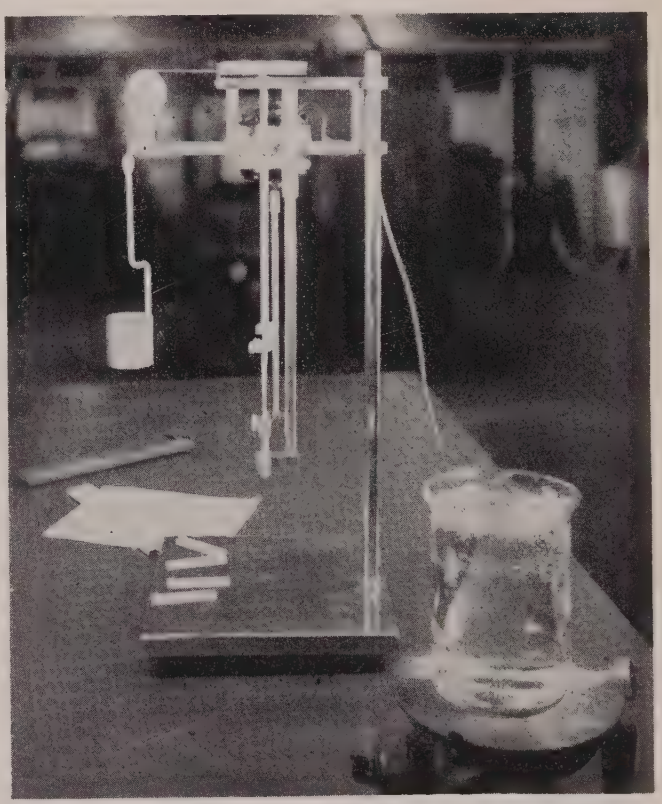

Fig. 50 – Torsional modulus of elasticity of part being measured at room temperature

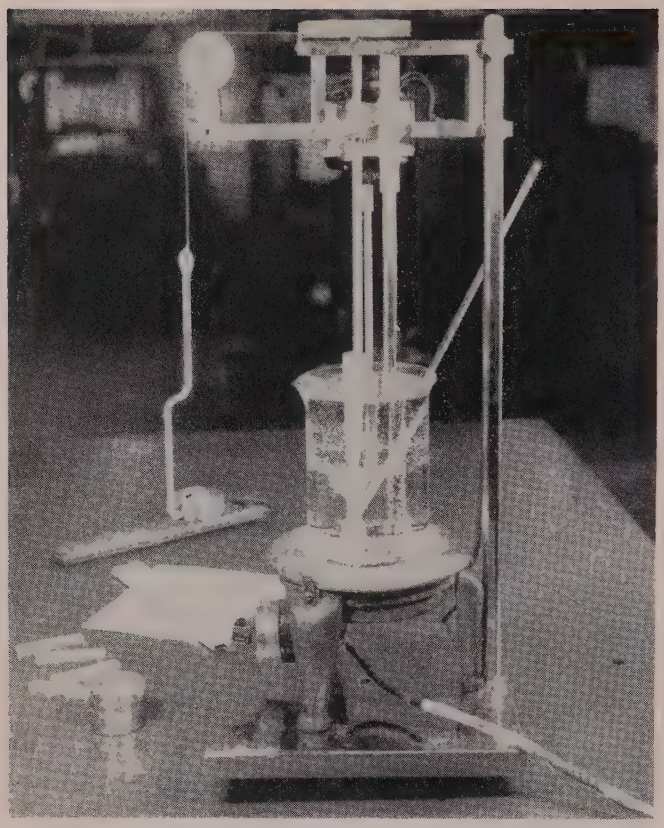

Fig. 51 – Torsional modulus of elasticity of part being measured at elevated temperature

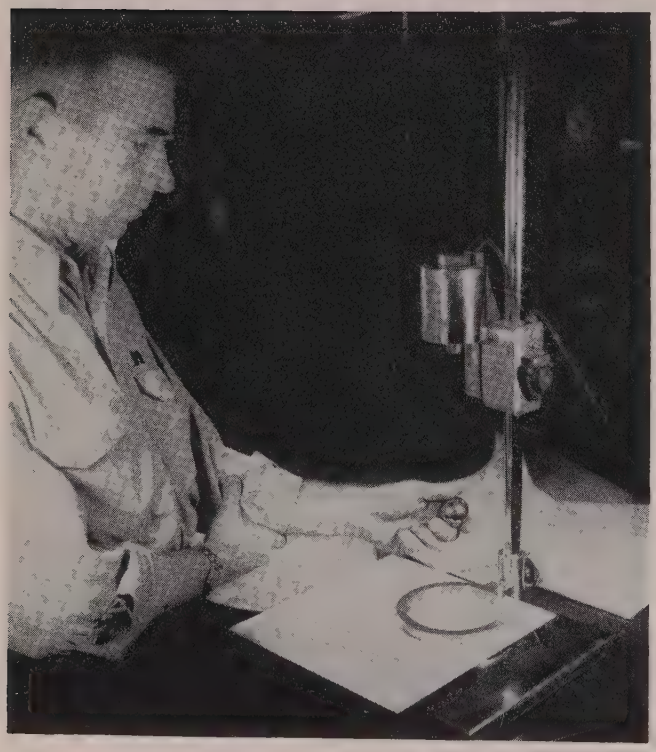

Fig. 52 – Equipment used to determine impact resistance

selected. This was chosen because it was the temperature of the paint baking ovens, and was not generally exceeded in actual use. A modulus of 400,000 psi minimum at this temperature has been established.

Equally important to the control of the rigidity of the material is control of the impact resistance, such as resistance to damage from stones thrown against the underside of the fender areas by the wheels. Here again, we reasoned that if the reinforced plastic material was equivalent to 0.036-in. thick steel it would be suitable.

We found that steel would dimple sufficiently to crack the paint if a 1½-in. diameter steel ball weighing ½ lb was dropped on the reverse side of the panel from a height of 7 in., but that it did not crack the paint when dropped 6 in. This was true at both room temperature and at –20 F. The polyester resin was found to lose about one-third of its room-temperature cracking resistance at –20 F, and so a 12-in. minimum drop of the ball without fracture was specified at room temperature, and an 8-in. drop was specified at –20 F.

We also require that a panel withstand a 10-in. drop after seven days of exposure to 212 F. The equipment used for determining the impact resistance of the steel and for testing the reinforced plastic material (Fig. 52) includes a 5-in. diameter circular framework, which is clamped against the test panel, a column graduated in inches, and a magnetic device for holding the steel ball.

After the ball has been dropped, the reverse side of the test sample is coated with a liquid dye penetrant, which is wiped off after being allowed to stand for several minutes. The test sample is then carefully examined, any indication of surface cracks being cause for rejection.

A hardness of Barcol 40 or 100 Rockwell M

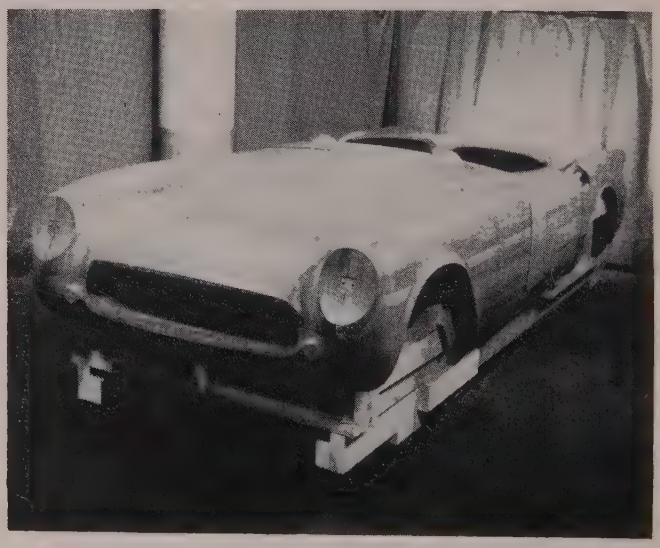

Fig. 53 – Individual wooden models fitted together, forming body

minimum is specified as a means of checking the completeness of cure of the resin. Material not completely cured will also show up on the other tests.

Water absorption of reinforced plastic is a fairly critical factor and should be kept as low as possible. Our allowance is 0.5% maximum by weight after 24-hr immersion at room temperature.

A surface finish for the exterior panels and the instrument panel of 100 microin. maximum to provide a surface suitable for finish painting is also specified.

Hand-Lay Process

The preliminary part of our body development program was carried on by our parts fabrication department at the Technical Center. The wood models made to check the surface development were constructed in such a manner as to be suitable for use in making the tools for production. After the individual wood models had been fitted together to check the overall surface development and the match of the panels to each other (Fig. 53), they were altered as necessary to provide for the various production tooling methods used.

The hand-lay tools used for the first production and experimental bodies were constructed to be somewhat stronger than usual because of the volume of parts that it was planned to make on them. In making these tools, a runout area (Fig. 54) is added around the edges of the wood model to allow extra stock for trimming. The model is coated with a parting agent and four layers of glass fiber cloth saturated with epoxy resins are laid up. After a 4-hr cure at room temperature, a coarse material is bonded with resin to the back of this laminate to provide a mechanical attachment for the 2 in. of reinforced plaster that is used as a base. The hand-lay mold (Fig. 55) is then attached to a hardwood frame by supports, which are bonded to the reinforced plaster. The completed unit is quite sturdy and durable.

In making a part such as the rear upper panel by the hand-lay process used at the Technical Center, the first step is to spray the female mold with a wax parting agent (Fig. 56) and allow it to dry for 5 min. A water-dispersed pva parting agent is sprayed on the waxed surface of the mold and allowed to dry approximately 10 min, forming a film on the surface.

Three gel coats of polyester resin (Fig. 57) are then applied to the mold surface. Between the second and third coats, circular wooden blocks are bolted to the mold at the wheel house opening to provide for a turned flange. After the third gel coat has been applied, and before it has hardened, lengths of glass fiber roving are cut and placed in the flange area to provide a reinforcement. While the gel coat is still tacky, glass fiber cloth is laid up in the mold, tailored to fit the contour, and smoothed out by hand, as shown in Fig. 58.

Fig. 54 – Wooden model with runout at edges

The process employs a sandwich construction with a layer of glass cloth on the outside and inside, and a layer of glass mat at the center. The resulting parts are somewhat stronger than when glass mat only is used, but the process is time-consuming and expensive and not suitable for any quantity production. A measured amount of poly-

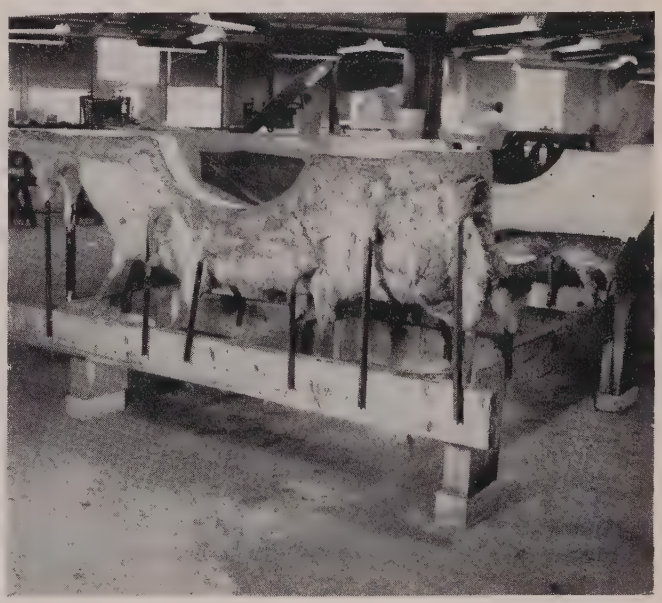

Fig. 55 – Hand-lay mold attached to hardwood frame

Fig. 56 – Female mold for rear upper panel being sprayed with parting agent

Fig. 57 – Gel coat of polyester resin being applied to mold surface

Fig. 58 – Glass fiber cloth being laid in mold

Fig. 59 – Part being removed from mold

ester resin to which the accelerator and promotor have been added is brushed on the glass cloth until it is saturated.

A rubber spatula is used in spreading the resin from the center of the cloth toward the outside and to remove all trapped air. The saturated cloth is allowed to cure for 30 to 45 min. A power grinder is then used to remove the rough spots, which would otherwise trap air when the balance of the laminate is applied. The layer of glass mat and the inner layer of cloth are then applied to the mold, the two layers are saturated with resin, and the laminate is allowed to cure for approximately 6 hr at room temperature.

After removing the flange forming blocks around the wheelhouse openings, the part is taken from the mold, as shown in Fig. 59. The flash is removed with a pneumatic saw and the edges are sanded. Water is used to remove the parting agent before assembling the part to the body.

Vacuum Bag Process

The vacuum bag tooling (Fig. 60) is made from a mahogany wood model to which a runout has been added to provide a surface for attaching hold-down clamps and extra stock length for the layup of the laminate. After the runout has been attached to the wood model, the surfaces are first sprayed with lacquer sealer to prevent the wood from absorbing moisture from the plaster cast that follows, and then coated with a wax parting agent to prevent adhesion of plaster.

A plaster and hemp mixture (Fig. 61) is prepared and spread 2 in. thick over the mahogany model to make a female plaster cast, which is allowed to harden at room temperature. A welded metal framework, normalized to relieve stress, is used as a base. It is lowered on the cast and the leg areas are attached with reinforced plaster.

After hardening, several coats of lacquer are applied to the finish surface of the cast to prevent chemical reaction with the resins used in making the mold. The lacquer surface (Fig. 62) is then sanded and polished and the unit is baked for 10 hr in an oven with a temperature of 120 F.

A steel frame is fabricated over the plaster cast, allowing approximately 2 in. between the framework and the cast for the resin mold. The frame is normalized and a wire mesh is attached to provide a base for the plastic.

A quick-setting putty coat prepared of phenolic resin, filler, and catalyst is troweled on the wire mesh to a thickness of approximately ¼ in. (Fig. 63) and baked in an oven for 6 hr at 180 F.

The form (Fig. 64) is then set on top of the female cast and the edges are sealed with reinforced plaster. A gate is made in the mold at the lowest point to provide an entrance for the liquid plastic, and vents are added at the high points of

Fig. 60 – Mahogany wood model used in vacuum bag process

Fig. 61 – Female plaster cast being formed

Fig. 62 – Mold surface being sanded and polished

Fig. 63 – Wire mesh covered with quick-setting putty

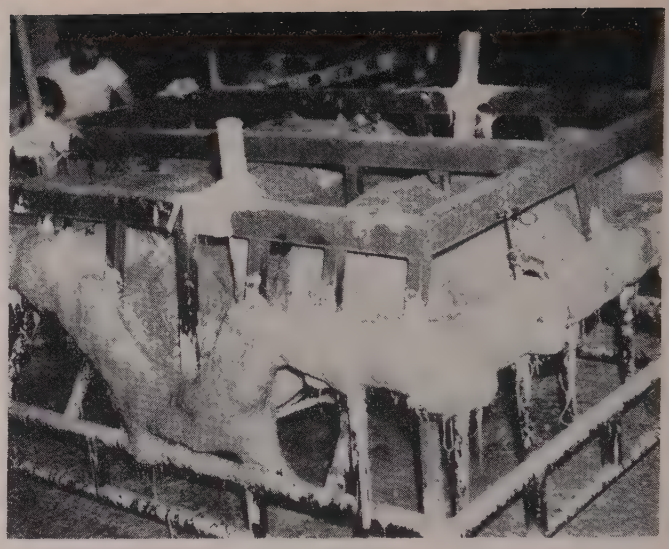

Fig. 64 – Form sealed in place on female cast

the mold to allow escape of air. A phenolic plastic mix (Fig. 65) is prepared and the mold is filled. Four to five hours are allowed to permit the resin to assume the shape of the mold and the air to escape through the vents.

After baking the entire unit in an oven at 180 F for 18 to 24 hr and allowing it to cool, the plaster is chipped away from the mated edges. The standpipes are removed and the lower surface of the mold is cleaned and coated with a sealer. The mold is then separated from the plaster cast and inverted for completion, as shown in Fig. 66. It is cleaned, trimmed, and buffed to a high luster to obtain a smooth finish on the parts made from the mold. It is then complete except for the addition of the vent holes, clamping ring, heating unit, and male plug, which is discussed later.

Adapting Vacuum Bag Method to Making Cor-

Fig. 65 – Mold is filled with phenolic plastic mix

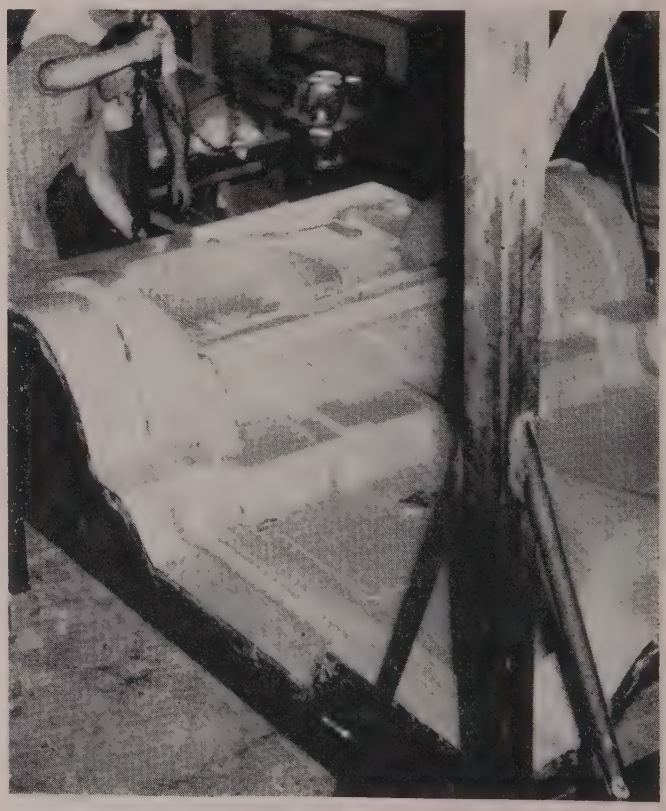

Fig. 66 – Mold inverted for finishing operations

vette Body Panels – In adapting the vacuum bag method to the making of panels for the first 300 Corvette bodies, many novel features were developed and very satisfactory parts produced.

After a hand coating of wax and a sprayed coating of water-emulsified pva have been applied as parting agents, three layers of 2-oz fiber glass mat are laid in the mold and the resin mixture is spread on, using ordinary paint brushes, as shown in Fig. 67.

Dry strips of mat are laid in strategic locations to aid the escape of trapped air, and metal vacuum troughs, which conduct the air out of the mold, are put in place. At various points around the mold, holes are provided under this vacuum trough. These holes are connected by tubing to a quart jar to trap excess resin.

A male plug, the shape of the inside of the part being made, is placed in position in the mold. This plug is made of layers of fiber glass and resin much like the plastic body panels and serves to provide a more uniform thickness of the finished part.

A pva sheet 0.003 in. thick, known as the bag, is spread over the top of the mold. It is sealed around the edges by means of a plastic tubing, approximately 1 in. in diameter, which has been taped to the upper frame. The frame (Fig. 68) is then ready to be dropped in place and clamped to the mold, after which the tubing is inflated with air and serves to tighten the clamps as well as to seal the bag. Vacuum is used to exhaust the air between the bag and the mold, resulting in an effective atmospheric pressure on the part of approximately 12 psi.

In addition to providing the means of sealing, the upper part of the mold includes a plywood box containing heat lamps, which are turned on to cure the part when the box is lowered in position. After a curing period of approximately 1 hr, the part is removed (Fig. 69) and forwarded to the trim department, where the flash is cut off and any surface defects sanded. The separator is carefully washed off before shipment to the assembly plant.

Fig. 67 – Resin mixture being spread on fiber glass mat laid in mold

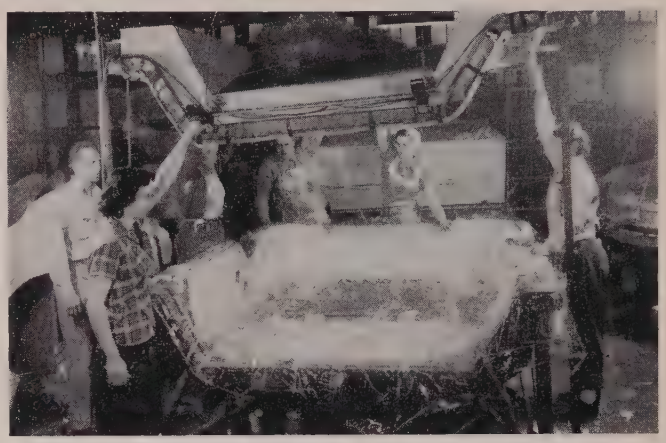

Fig. 68 – Frame ready to be dropped in place and clamped to mold

Matched Metal Die Process

The matched metal die tooling also begins with the wood model to which a flange runout has been attached. The model is sprayed with lacquer sealer to prevent absorption of moisture, and is coated with a wax parting agent.

The four major units made to facilitate the machining of the metal dies are:

1. A Keller model for the female die.
2. A spotting rack for the female die.
3. A Keller model for the male die.
4. A spotting rack for the male die.

The female Keller model (Fig. 70) is made directly from the wood model. The waxed surface of the wood model is sprayed with a coat of epoxy resin, and allowed to stand until tacky. A layer of

Fig. 69 – Panel after removal from mold

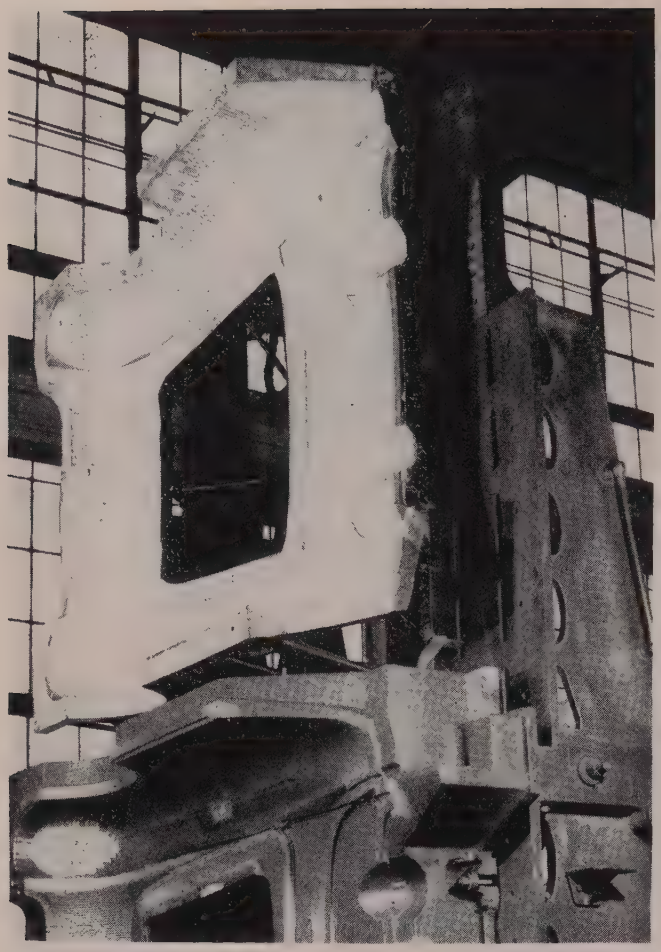

Fig. 70 – Female Keller model

glass fiber cloth is applied and smoothed out to the contour of the form. Fifteen to twenty layers of glass cloth are placed on a bench, saturated with epoxy resin, and laid-up over the mahogany model.

A normalized steel framework is constructed and lowered over the laminate while it is still tacky. The legs are tied in with glass cloth that has been saturated with resin, and the assembled unit is allowed to cure for 4-6 hr.

The female Keller model is coated with a parting agent and 15-20 layers of glass cloth saturated with resin are applied to make a spotting rack. A steel framework is attached to this layup in the same manner as was used for the female Keller model. This spotting rack (Fig. 71) is used for barbering the female die after it has been kellered. Fiber glass reinforced phenolic tubing is also used in place of the steel framework by some manufacturers and is claimed to be less subject to dimensional variations. However, there is a difference of opinion within the industry regarding the relative merits of these two methods.

The male Keller model is made from the female Keller model. Sheet wax, as thick as the part to be produced from the tools, is now laid up in the female Keller model, stopping at the trim line, as shown in Fig. 72. Heat is used as required to shape

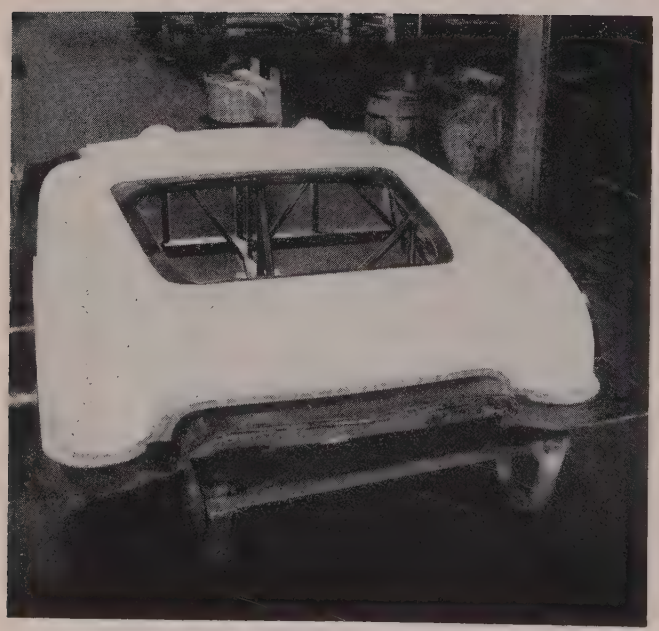

Fig. 71 – Spotting rack

Fig. 72 – Sheet wax for part thickness being placed in female Keller model

the wax to the contour of the mold, and a parting agent is applied to the wax-lined surface.

The male Keller model is then laid up in this unit and a framework is attached in the same manner as used in making the female Keller model. Fig. 73 shows this model in position on a Keller machine.

The spotting rack for the male die (Fig. 74) is made directly from the male Keller model in the same manner as was used in making the spotting rack for the female die. The use of the female spotting rack is, of course, also to facilitate the barbering of the male die.

Figs. 75 and 76 show a typical rough casting used for matched metal dies. The casting is cored out to provide for the passage of the steam used as the heating medium in curing the reinforced plastic parts. A metal plate bolts to the bottom of this casting to complete the enclosure.

The kellering of these dies follows the usual practice in making tools for our pressed metal parts (Fig. 77), the main difference being that only one die is required to make the part of reinforced plastic instead of the many dies used in making an equivalent steel part.

As removed from the Keller machine, the die surface is rough and requires hand grinding to obtain the final shape. This process is known as barbering the die and it is at this point that the spotting rack comes into its own. Through use of the spotting rack, it is possible to finish the male or female die completely without having the counterpart available. After the dies have been barbered, the pinch-off is fitted to within 0.001-0.002 in. and flame hardened. The die tryout procedure then provides the final touches.

New Plant Built to Manufacture Corvettes – A 30,000 sq ft plant was constructed to make parts for the 10,000 Corvettes to be produced during 1954. This is the first plant in existence built primarily to make plastic automobile body parts.

Fifteen huge presses (Fig. 78) ranging in capacity from 250 to 500 tons were installed in the pit line. These presses have beds ranging from 42 x 49

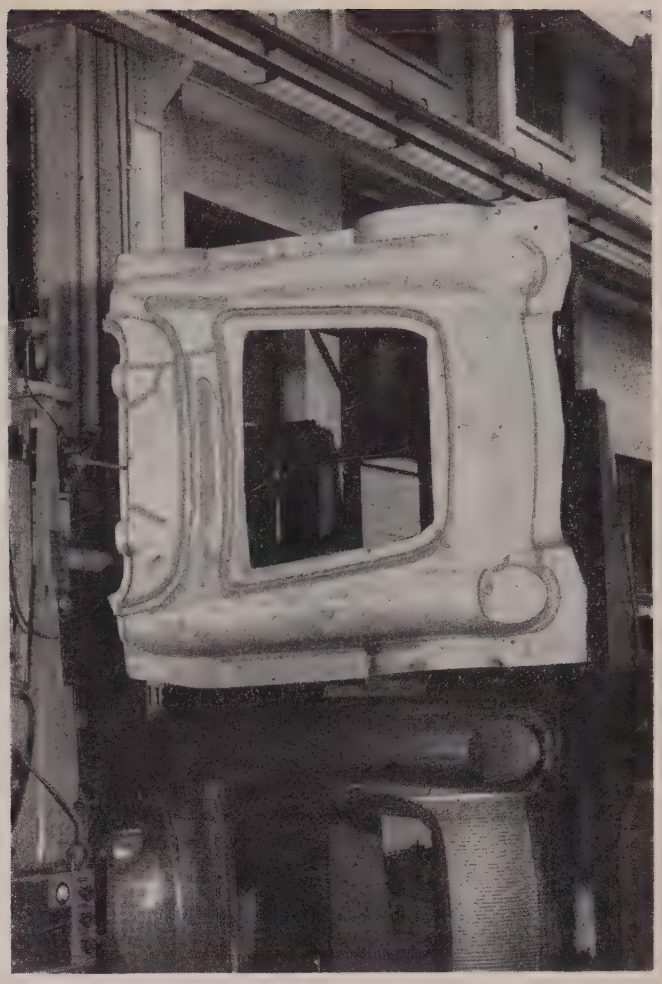

Fig. 73 – Male Keller model in position on Keller machine

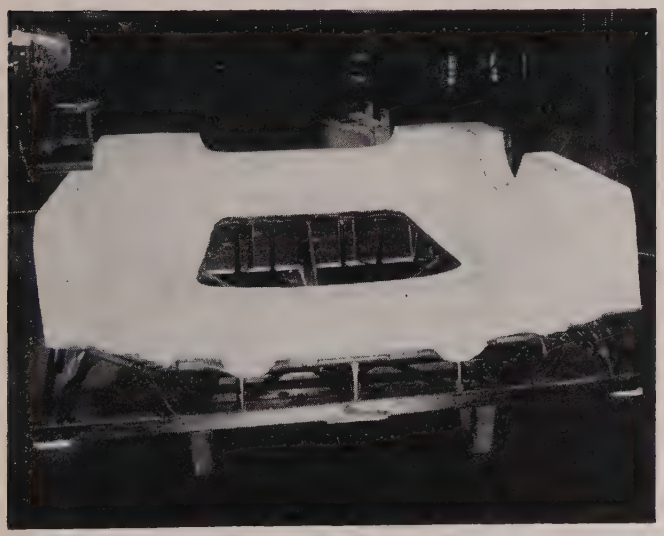

Fig. 74 – Spotting rack for male die

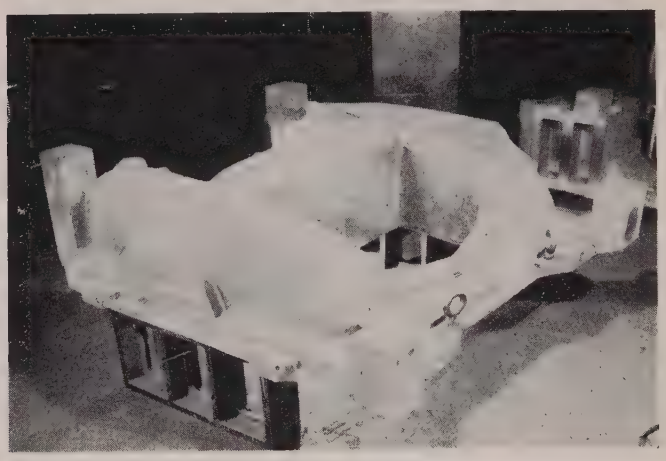

Fig. 75 – Top view of rough casting used for matched metal dies

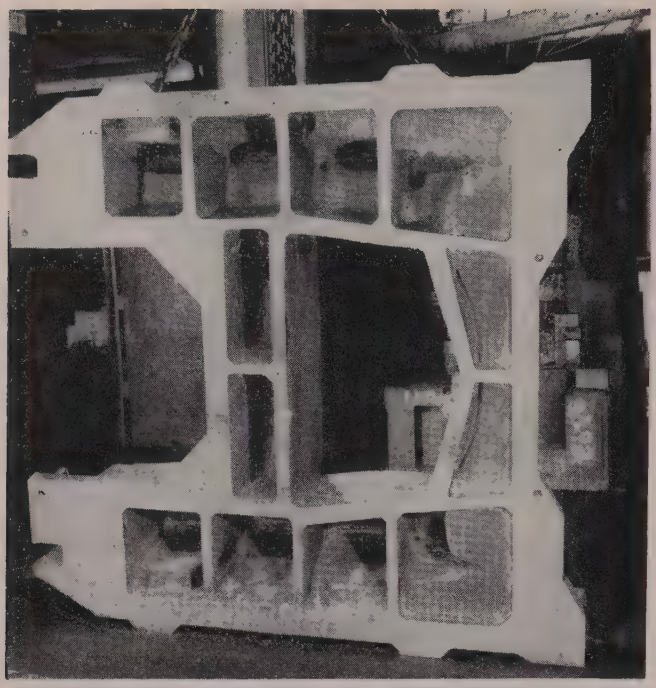

Fig. 76 – Underneath view of rough casting used for matched metal dies

Fig. 77 – Die being kellered

to 84 x 144 in. in size, the largest being for the underbody. This part is 120 in. long, 72 in. wide, and 26 in. deep. In addition, there are 10 presses of 18-ton capacity with 20 in. square beds. The hydraulic line pressure used is 2000 psi.

When completed, the plant will be equipped with preform machines, conveyer systems, ovens, and a finishing department compatible with the press capacity. It will be capable of producing 10 million pounds of reinforced plastic parts per year. The process used in this new plant is similar to that previously described; preforms are made in the same manner (Fig. 79), but are much larger than ever produced before.

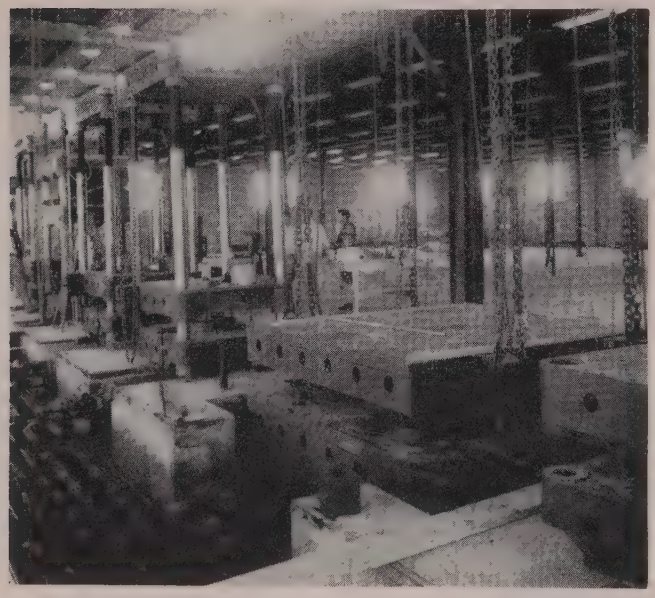

Fig. 78 – Presses used to make plastic body parts

The preform and screen are placed on a conveyer (Fig. 80) and passed through an oven to cure the resin that has been used to hold the preform together. The temperature in the oven is 350 to 400 F and the part takes 3 min to pass through.

In the vacuum bag and hand-lay processes, gel coat is used for all exterior parts requiring a smooth surface. This method is not suitable for use with matched metal dies because of the time element involved and it was, therefore, necessary to develop some other means. The problem was finally solved by a light-weight, fine-textured surface mat (Fig. 81) made up of extremely fine filaments of fiber glass. It would take 204 of these filaments to equal one strand of the conventional 2-oz mat used in the hand-lay process. The weight of this material is only 4 oz per sq yd.

This surface mat, tailored to shape, is laid in the metal die. The preform is added and a measured amount of resin mixture is poured on it, as shown in Fig. 82. The die is closed for approximately 3 min, after which it automatically opens and the part is removed, as shown in Fig. 83. The edges of the parts are then filed or sanded to remove fiber glass that was not completely pinched off by the die and visual inspection is made for cracks, low spots, and other imperfections.

Cracked parts are discarded, rough spots sanded, and small pits are filled with resin. The small repaired areas are cured by heat lamps and, if found

Fig. 79 – Preform being made

Fig. 81 – Fine-textured surface mat being laid in die to provide smooth finish to part

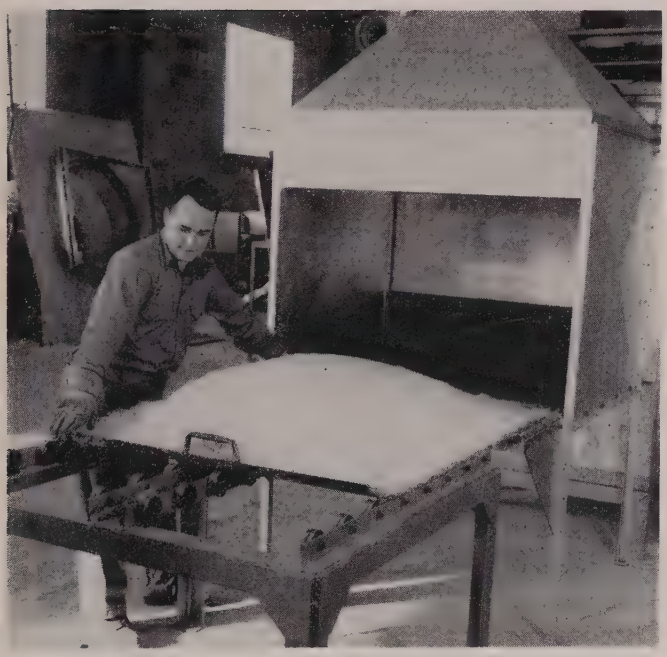

Fig. 80 – Preform and screen being placed on conveyer, which will pass it through oven

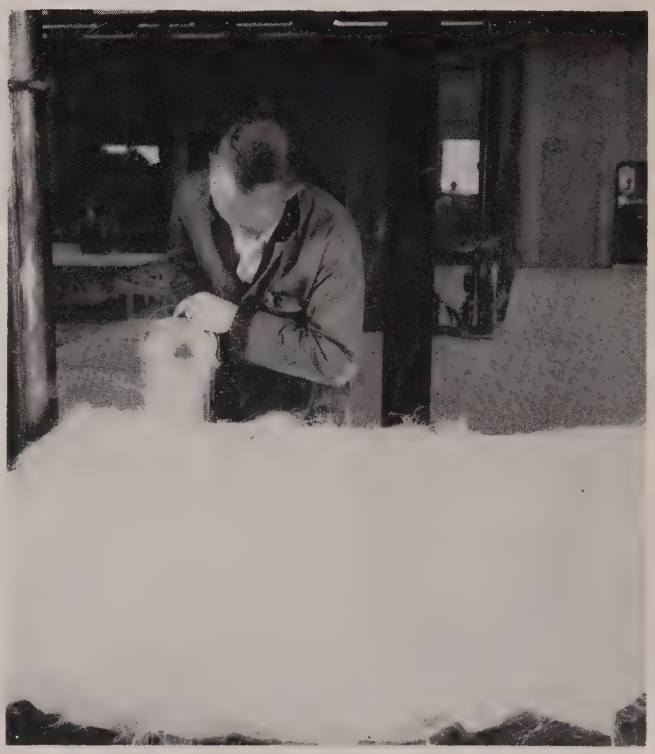

Fig. 82 – Resin mixture being poured on preform

Fig. 83 – Part being removed from die

satisfactory, the parts are rubber stamped as O.K. and shipped to the assembly plant. The operation of removing a parting agent is unnecessary with this process, since a coating of wax applied to the die once or twice a day is all that is necessary to keep the parts from adhering.

Small parts are boxed for shipment and larger parts are simply stacked on pallets.

Body Assembly

The assembling of the first 300 Corvette bodies was done at the Chevrolet Flint Assembly Plant, and the larger quantity of 10,000 units in 1954 is being produced at the St. Louis Assembly Plant.

The reinforced plastic parts are received in various stages of completion. Some detail parts are complete with attaching holes; others require holes to be added; and some trimming operations are involved. Many of them, such as the hood, top stowage compartment, and luggage compartment covers, and the side doors are received in a subassembly condition. These parts are stored on pallets and in racks at strategic locations along the assembly line. In general, parts designed for the matched metal die process nest very well and do not require excessive storage space, as shown in Fig. 84.

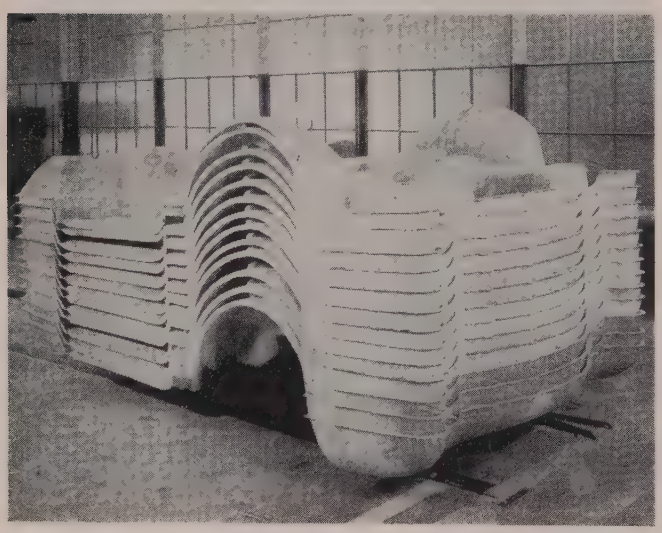

Fig. 84 – Nest of reinforced plastic parts

The assembling operation starts with the underbody, which is the largest one-piece panel used. It is placed in a holding fixture near the beginning of the assembly line and securely clamped. A drill jig containing bushings for drilling the holes is lowered by an electric hoist (Fig. 85) and the required holes are drilled, as shown in Fig. 86. This jig is made up entirely of reinforced phenolic tubing, and the locating faces are made of fiber glass reinforced epoxy or polyester resins.

The underbody is then transferred to a rotating-type assembling fixture where body sills and floor reinforcements are bonded and riveted in place, as shown in Fig. 87. Aluminum rivets 3/16 in. in diameter are used, primarily to avoid the necessity of clamping fixtures to hold the parts together until the bonding resin is cured. In this manner, the assembling sequence can proceed without delay. Actually, after the bond is cured the rivets are unnecessary. The bonding material used is a mixture of activated polyester resin and chopped glass fiber. It is necessary to disc-grind the mating surfaces to assure a good attachment.

The assembled underbody is then placed on a body truck (Fig. 88), which serves both as a conveyer and assembling jig. This unit is very substantially constructed on a welded-steel box-section base and holds the body in shape during the assembling sequence. The absence of overhead fixtures and cumbersome spot-welding equipment on the Corvette body assembly line is quite noticeable.

The rear upper panel assembly (Fig. 89), consisting of the upper panel, the gasoline tank filler box, and the bulkhead is built upon a feeder line. As in the case of the underbody assembly, and throughout the entire operation, the joints are bonded and riveted together. This assembly is placed on the underbody (Fig. 90) reworking the flanges as necessary to obtain a good fit before bonding. Very little of this type of work is required on parts made by the matched metal die process. Holes are drilled through the flanges for the rivets at this time, and the parts are held together during the riveting operation by use of spring fasteners.

The front upper panel assembly also is built up on a feeder line and starts with the front fender skirts and the radiator support. The plastic parts are bonded and riveted together and the metal radiator support, which includes cross brace rods,

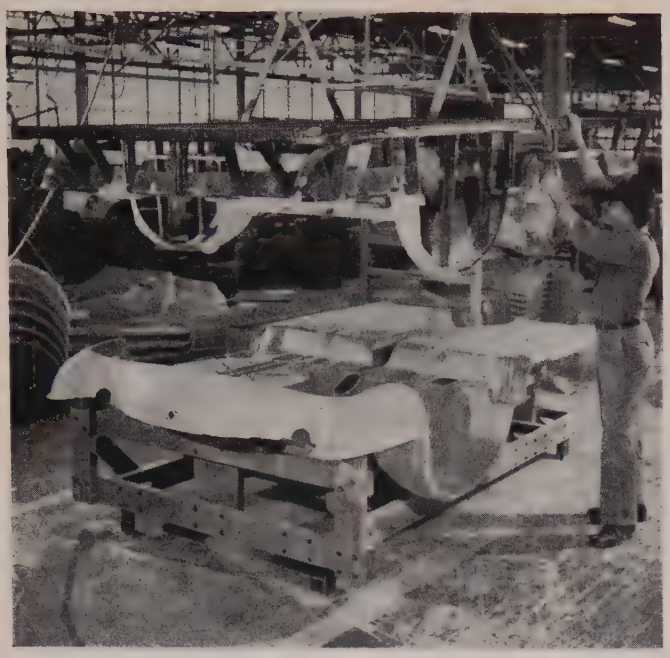

Fig. 85 – Body assembly operation starts with placing of underbody in holding fixture

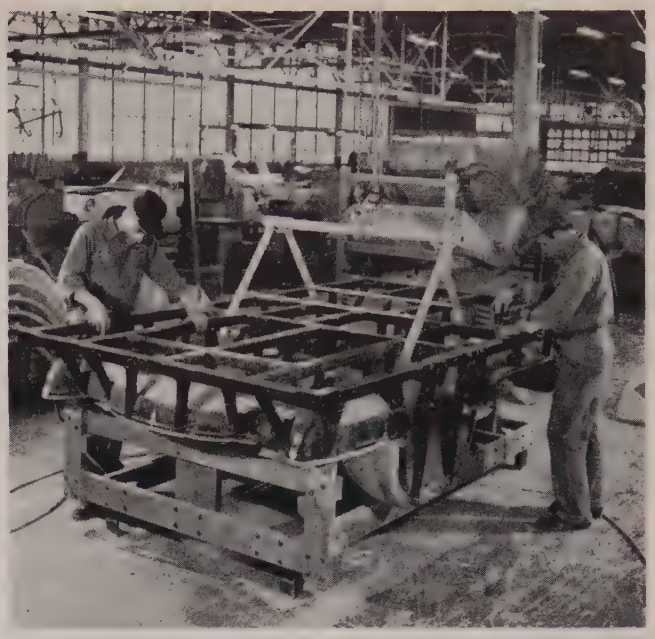

Fig. 86 – After drill jig is in place, required holes are drilled

Fig. 87 – Body sills and floor reinforcements being bonded and riveted to underbody

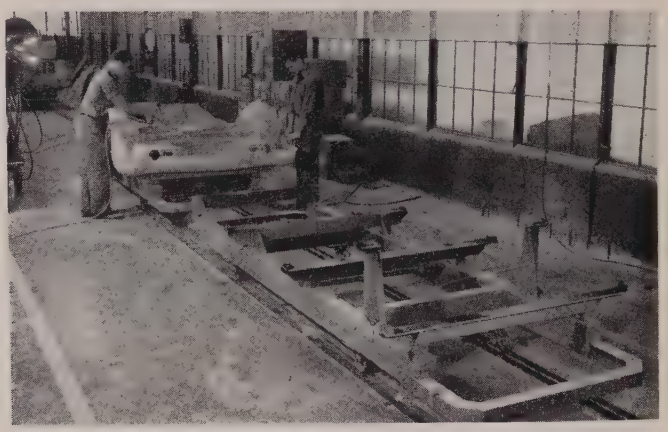

Fig. 88 – Assembled underbody on body truck

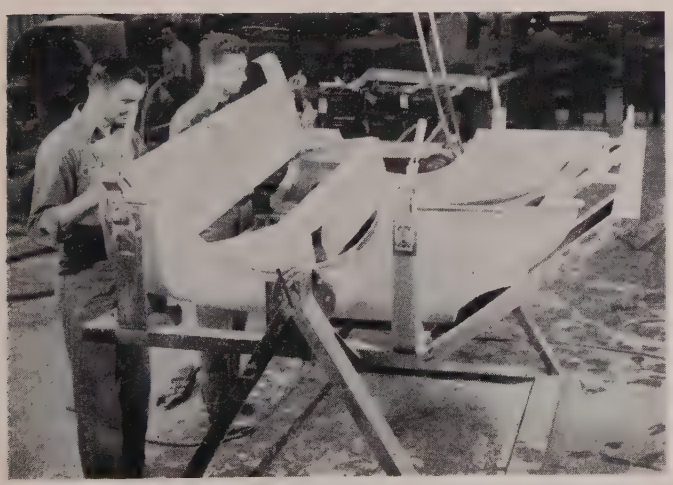

Fig. 89 – Rear upper panel being subassembled on feeder line

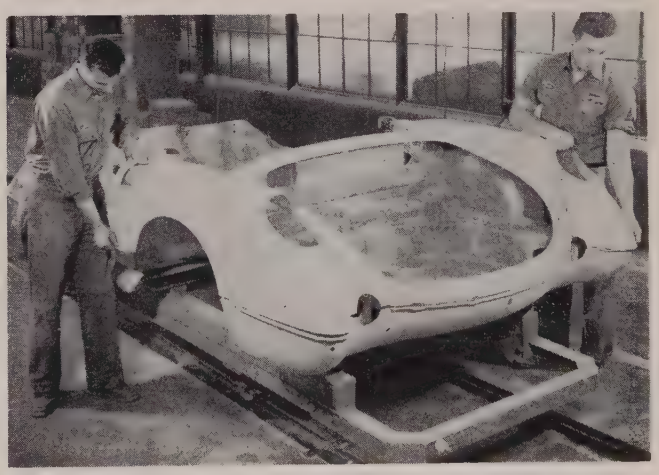

Fig. 90 – Rear upper panel assembly being fitted on underbody

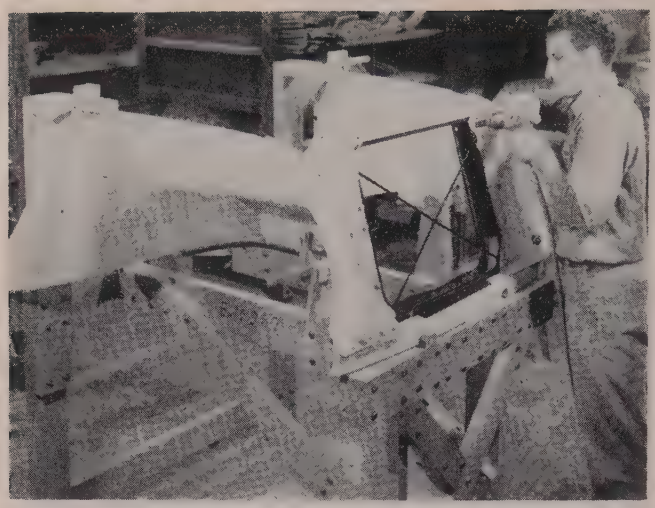

Fig. 91 – Front fender skirt assembly being built on feeder line

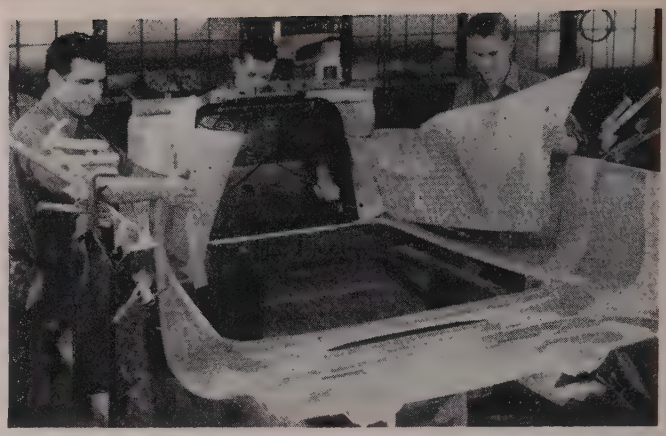

Fig. 92 – Front fender skirt assembly being added to front upper panel

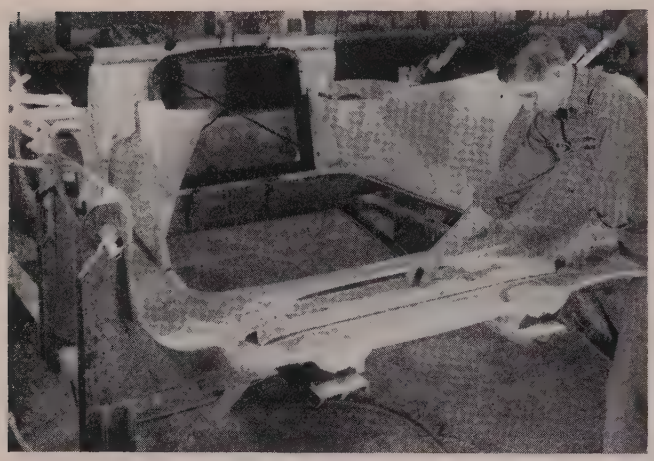

Fig. 93 – Cowl bar reinforcement being added to front upper panel

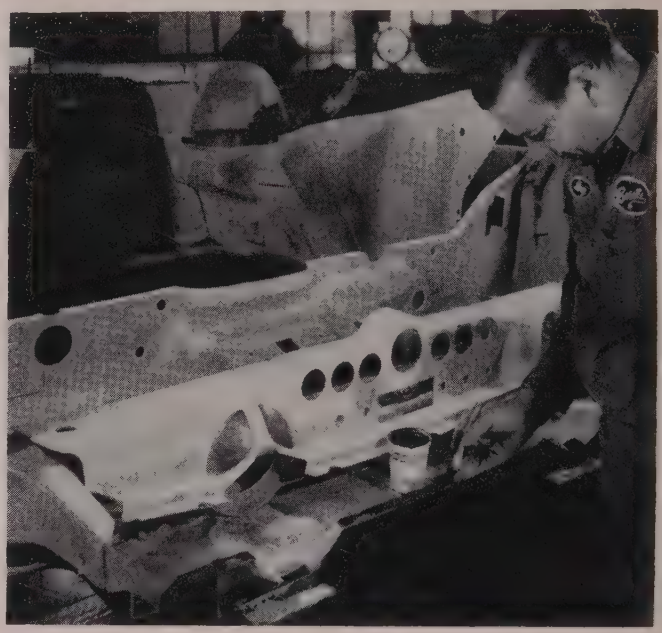

Fig. 94 – Putting adhesive on front upper panel for bonding of instrument panel

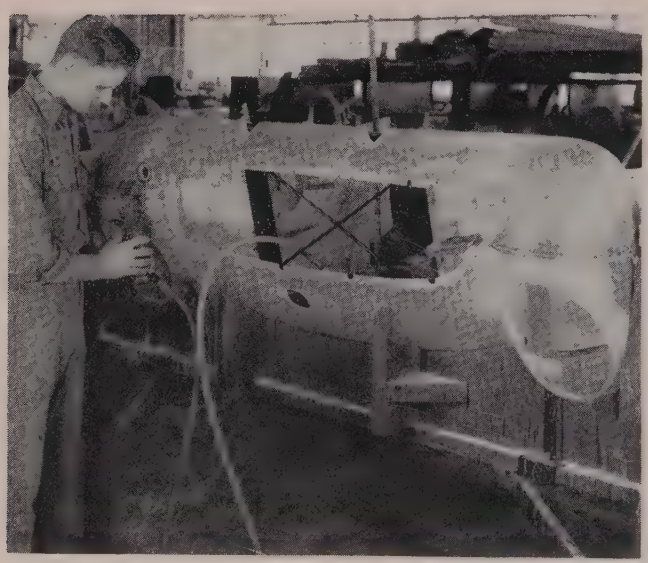

Fig. 95 – Front lower panel being assembled to front upper panel

Fig. 96 – Completed front upper panel assembly being positioned on underbody in main assembling fixture

is bolted in place, as shown in Fig. 91. This assembly (Fig. 92) is added to the front upper panel in an inverted position in the next fixture. It is essential that parts be clamped securely in the correct shape during the assembling operations because, once they are bonded together, a very rigid assembly results, which cannot be twisted into shape, as is the case with steel parts.

Anchor plates for windshield wiper transmissions are riveted to the cowl bar reinforcement, which is assembled to the upper panel, as shown in Fig. 93. The riveting of steel anchor nuts and reinforcements to the plastic to provide hard spots for attachment of body hardware is common practice throughout this body.

The upper portion of the dash panel, the hinge pillars, and the instrument panel lower portion are then assembled to the front upper panel.

The top portion of the instrument panel is included in the front upper panel and a flange provided at the rear edge of this surface is used for attachment of the lower portion of the instrument panel. This joint is bonded (Fig. 94) and riveted and covered later with a trim pad to provide a finished appearance. Following this, the front lower panel is assembled. This completes the opening for the radiator grille (Fig. 95) and also provides the lower portions of the front fenders.

There is a conspicuous absence of any positioning fixture. Once the manufacturing tools are tuned up, the parts made are very uniform. There is no springback in reinforced plastic parts and the flanges match together so well that fixtures are unnecessary.

The completed front upper panel assembly is positioned on the main assembling fixture, as shown in Fig. 96. It is located at the front by the radiator support mounting holes and at the rear by the dash portion of the underbody. No cumbersome door gates or fixtures are required. The door opening size is checked and the assembly is bonded and riveted to the underbody.

The rocker side panels (Fig. 97) are next clamped in place. These panels provide the outer portion of the body sill box section and the lower portion of the body side between the front and rear wheel openings. The mated flanges are drilled, bonded, and riveted. The small clamps seen in Fig. 98 are operated by an ordinary spring fastener plier.

Tail light and head light housings, tail pipe ducts, and seat separator panels are next assembled to the body. Side door assemblies (Fig. 99) and the hood, deck lid, and top compartment cover assemblies are fitted to the body and then removed for painting. Miscellaneous holes for the assembling of interior trim moldings and body hardware, not previously provided in the pieces, are now jig drilled and the body is ready for the finishing operations.

Before painting, all surfaces are cleaned to remove any parting agent that might have been missed when the panels were cleaned at the fabricating plant, and the surfaces requiring a high-luster paint finish are sanded, as shown in Fig. 100. The body is then steam cleaned (Fig. 101), wiped with denatured alcohol, and the remaining dust removed with a varnish tack rag. Body glaze is used to cover any imperfections and the body is

Fig. 97 – Rocker side panels being positioned for assembly

Fig. 98 – Mated flanges being riveted

Fig. 99 – Side door assembly being fitted to body

baked, allowed to cool, and the excess body glaze sanded off.

The painting procedure that follows is the same as that employed for a steel body. Two surfacing coats are applied, baked, and finish sanded. Two

Fig. 101 – Body being steam cleaned

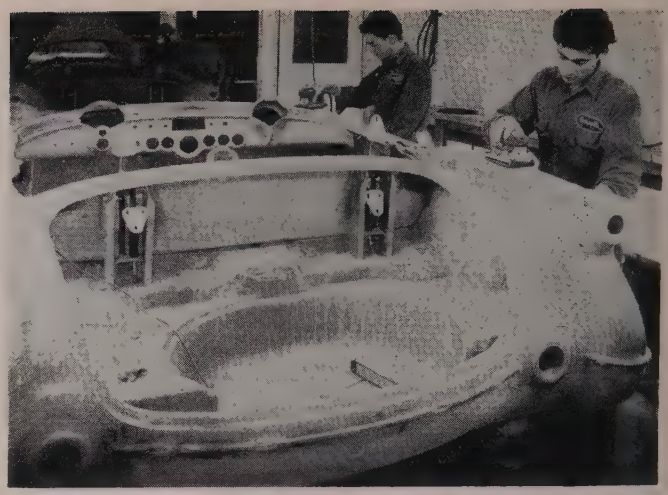

Fig. 100 – Body surfaces being sanded, prior to painting

lap coats of finish lacquer are sprayed on all inner exposed surfaces and three lap coats of finish lacquer are sprayed over the exterior surfaces and the instrument panel. All paint coats are baked at an oven temperature of 180 F.

The engine and luggage compartments are then sprayed (Fig. 102), using masks made of reinforced plastic. The finished surfaces are water sanded, as required, and polished in the same manner as used for steel bodies.

The body then moves down the trim assembly line, where it is completed (Fig. 103), including the side doors, top stowage compartment cover, and deck lid, which were fitted previously. It is then dropped on the chassis, bolted in place, and the hood installed and adjusted.

This completes the assembling of a Corvette and, after inspection, a check of performance is conducted and necessary adjustments made before delivery to the dealer.

The Future

The tremendous interest aroused by the Chevrolet Corvette wherever it has been shown leaves no doubt as to the public's desire to be able to buy an American-built sports car.

By far the greatest part of this interest has centered on the reinforced plastic body. This is not only because of its novelty and its appeal as a natural styling medium but because of the obvious advantages it has in overall sports car design.

The Corvette has ideal weight distribution and a low center of gravity. Extremely stable and safe to drive, the car combines excellent acceleration and cornering ability with thoroughly American standards of comfort and convenience.

It must be remembered that, promising as it is, this new industry of reinforced plastics is in its infancy.

Fig. 102 – Luggage compartment being masked for painting

Fig. 103 – Final body assembly operations before mounting on chassis

There is far more to be done in the development of material specifications and related testing. Advances in technology will provide us with new laminates of greater strength, lightness of weight, and with even more durability than we are able to produce at present. So far, we have barely scratched the surface in exploring the possibilities of reinforced plastic materials, and it seems certain that their development will profoundly influence future body design.

The whole history of the automobile industry is one of challenge to man's inventive genius. And, whatever the future role of reinforced plastics in the automotive field, the story of the Corvette body is, I believe, thoroughly in keeping with the tradition of continually finding new ways to do new things.

Fig. 104 – Damaged panel before repairs

APPENDIX

Repair Technique

The use of reinforced plastic for a production automobile body required the development of adequate procedures for repairing any type of damage that might occur. Damage to a reinforced plastic body tends to be localized, while repair procedures are simple and require no special skills.

Through tests, as well as from field experience to date, collision damage to reinforced plastic parts has proved to be of three basic types:

1. Cracks.
2. Small, repairable holes.
3. Extensive damage requiring the insertion of replacement panels.

Small cracks are repaired by first sanding the

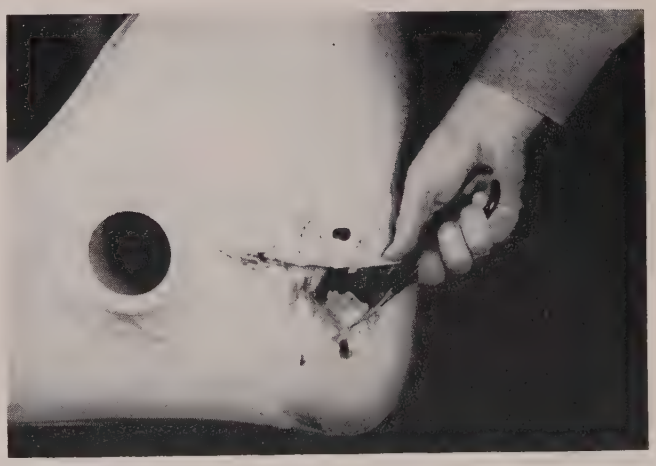

Fig. 105 – Loose and damaged material being cut away

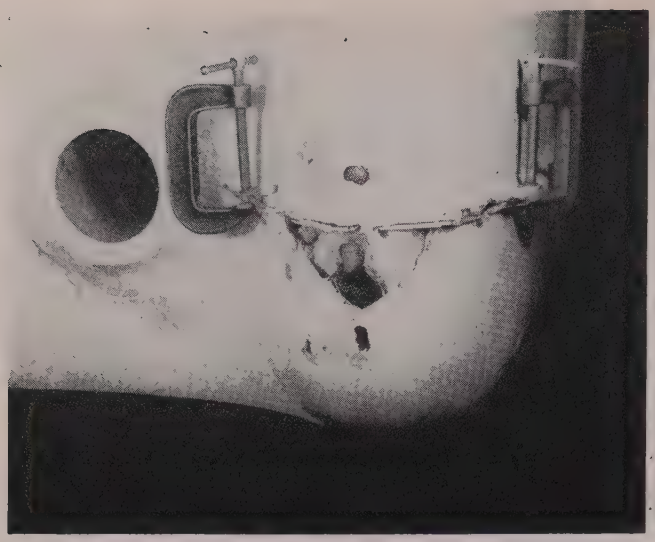

Fig. 106 – Resin saturated fiber glass cloth has been applied to rear surface for backup

Fig. 107 – Prepared resin being spread to fill in cavity on patched surface

Fig. 108 – Removing badly damaged portion of panel for replacement

Fig. 109 – Damaged section has been hack-sawed out

Fig. 110 – Repair panel section secured in place

Fig. 111 – Repaired surface ready for painting

damaged area to provide good adhesion of the repair resin. After sanding, a resin mixture containing chopped glass fibers is prepared and the crack filled. When the resin has cured, it is sanded and finished to the contour of the part, which is then ready for painting.

When more severe damage has occurred and the panel is fractured with a break-through (Fig. 104), the loose material is cut away to the sound surface, as shown in Fig. 105. The under side of the damaged surface is cleaned and sanded and a patch of glass fiber cloth wetted with resin is applied, as shown in Fig. 106. After the patched surface has hardened, the outside surface is sanded and prepared resin is spread on (Fig. 107), filling the hole to the contour of the surrounding surface. This is allowed to harden and is then sanded for the conventional painting operations that follow.

If large breaks have occurred, a replacement panel is required. The damaged section of the body is hack-sawed out (Figs. 108 and 109), and the sawed edge is beveled.

The mating edge of the repair panel section is also beveled, forming a V for the resin filler. A repair panel is secured in place (Fig. 110) and reinforcing strips of resin-saturated glass cloth are applied to the under side at the joint. The V in the outer surface is filled with resin mixture. After the resin has hardened, it is sanded to the contour of the surrounding surface (Fig. 111) and painted.

DISCUSSION

How to Select Fiber Glass Mat

– J. S. Lunn
Lunn Laminates, Inc.

THERE are many standard types of mat, and special types can be purchased when the quantity justifies the run. Mats are available in several weights and are graded in ounces per square foot. Standard weights run from ¾ to 6 oz per sq ft.

Mats are available with a wide variety of binders to hold them together, although some mat has no binder at all, being held together by physical means such as "needling." Various formulations of binders are used, based on polyesters, melamines, ureas, polystyrenes, starch, and the like. Binders are selected to impart qualities to the mat such as formability, color, wash resistance, and varying degrees of solubility to the laminating resin to be used.

Mat is available with varying lengths of glass fibers, from very short to continuous lengths of fiber or combinations of lengths. Proper selection imparts desired properties to the mat. Mat can be designed to give strength in certain directions by incorporating continuous fibers or by controlling the orientation of fibers in the mat.

The first consideration in choosing a mat for a job is, of course, to select a mat that will produce the properties desired in the ultimate molding. For example, in choosing a mat for a decorative lampshade the primary consideration is appearance of the finished laminate. An expanded finishing type of mat with perhaps a starch binder might be the choice. Its low strength would be unimportant. For bag molding an automobile part, strength and impact resistance are primary and appearance secondary.

Having determined what standard or special types of mat will do the job to be done, the next consideration is to choose a mat that will be easy to handle and mold in the particular application. The points to consider are the ability of the mat to form to the contours to be molded, ease of cutting, resistance to washing, ease of tearing. In the Corvette program we chose for bag molding the body parts a mat that has no chemical binder, but is held together by mechanical means. It forms nicely to the contours of the mold. It is easily cut into patterns by portable electric knives and the edges of the patterns can be torn where they overlap with other patterns, which minimizes the effect of overlap thickness. Washing was a minor consideration because in our bag molding process we apply the resin equally over the whole surface of the mat, and the resin, therefore, does not have to flow much during the molding operation. This mat would be entirely unsuitable in a contact molding operation since the needling, which holds it together, tends to hold layers of the mat apart when no pressure is used. It is also relatively unsuitable in bag molding where no match mold is used, as it tends to move around under pressure, and produces a laminate of varying wall thickness.

Development of Corvette Body Brought Some Unusual Problems

– Carl C. Jakust
Chevrolet Motor Division, GMC

ONE of the things we didn't think about right at the beginning, but later discovered we had to do, was to use a double-wire electrical system all through the body. The battery was grounded to the engine and, of course, the positive cable attached directly to the starter, and the chassis frame was not used at all as a conductor.

Another item that developed was the need for a good ignition suppression system in order to eliminate static in the radio of the Corvette, as well as in the radios of other cars. The very factor that made fiber glass an ideal material for radomes made it undesirable from the standpoint of ignition interference with radio reception.

We also ran across a couple of minor legal problems in connection with the screens over the headlamps and the rear license plate compartment. In these cases, the trouble wasn't entirely unexpected but we elected to continue with the screens over the headlamps and the plexiglass over the license plate because of the styling advantages. In order to overcome the possibility of certain states refusing to license cars with these features, we made these parts so that they could be easily removed.

I believe the biggest unexpected trouble we ran into was water. It wasn't too long after the first model came off the line that we got the first reports of this trouble from no less a person than E. N. Cole, our chief engineer. He had taken one of the first Corvettes on a trip to determine how good a job the engineers had done.

Unfortunately, it rained and the side door seals, which were located between the top edges of the doors and the cowl and which were intended to conduct the water forward

to run down the front edges of the door, leaked at the rear ends and conducted all the water that ran off the windshield into the interior of the body. This process was assisted by air pressure, which could not have been improved on as a means of accelerating the entrance of the water.

He soon found out that reinforced plastic made a pretty good boat because the water started to collect in the floor and before long there was quite a pool. The additional moisture inside the body caused the windows to steam up and when he reached into the door storage compartment to get a rag to wipe the windshield off—he found the water level in the door even higher than that in the floor. In fact, there was 4 in. of water in the door and, to top it off, his camera was in it.

I won't go into the details of the instructions we received to correct this deficiency, but can assure you that the action was prompt. We improved the seal between the door and the pillar and—just in case there was still a leak—added holes in the bottom of the side doors and in the floor of the cockpit.

How Resin Manufacturers Are Developing a Suitable Resin

— R. L. Reid
Bakelite Co.

I HAVE been asked the question, "What are the resin suppliers doing to meet the needs of the fabricator for a suitable resin?" Traditionally, automotive engineers have been trained to think in terms of conventional engineering materials, particularly steel. The application of reinforced plastics on the Corvette introduced new problems to Chevrolet, the fabricator, and the resin suppliers.

When the Chevrolet engineers designed the Corvette, they used their experience and background to set up the basic standards for performance. The fabricators were faced with the problems of matching or surpassing the conventional materials in regard to strength, appearance, durability, and many other factors. At this point, it was realized that much more information on design data was required than was immediately available.

To start out with, the resin suppliers first brought samples of their products to the fabricator's plant to match metal mold test panels. These test panels were evaluated to obtain basic and initial strength properties. At a later date, 11 different resin companies met at the fabricator's plant to have a resin evaluation. Each company molded 20 identical parts from actual body production. These parts were distributed and evaluated by the same 11 resin suppliers along with two fiber glass companies and the fabricator. The purpose of this evaluation was to determine, according to Chevrolet strength specifications, which were the best resins to use for the job.

A special problem, that of reverse impact, which can be illustrated by a stone hitting the underside of a body part, and showing effect on the reverse or outside painted part by resin separating from the glass, was something new to the resin suppliers. For this requirement, new resins or blends had to be supplied to the fabricator. The successful solution of this problem may be attributed to the close cooperation of the resin suppliers with the fiber glass companies.

Speed of production was a primary consideration, and had to be coordinated with good moldability and excellent surface finish.

At this time the initial requirements have been met and exceeded. Resin companies are conducting studies designed to reduce production cycles to at least as fast or faster than now required for steel stampings. Work is continually going on to improve the strength of reinforced plastics, both from a mechanical and chemical viewpoint.

Polyesters to resin manufacturers are not new. It is true that a major use of polyester resins was developed during the Second World War. However, all resin and fiber glass manufacturers have been aware for many years of the outstanding physical properties, which emphasize the high strength and corrosion resistance obtainable from reinforced plastics.

Pressure Bag Method Has Many Advantages

— I. M. Scott
Winner Mfg. Co.

ALTHOUGH matched metal dies constitute the lowest cost method of producing reinforced plastic panels, this is true only if:

(a) Volume is sufficient to warrant cost of matched metal die tooling.

(b) Part is small enough to be produced on standard press equipment at normal hydraulic pressures.

(c) Part is designed without undercuts or reverse draft. (Split molds generally are not practicable.)

Recognizing these conditions, we find that the advantages of using the pressure bag method are realized when:

(a) Parts are too large for standard press and preform equipment.

(b) Parts have undercuts or reverse draft.

(c) Parts have inserts and buildups, since such parts are not suitable for press molding due to the lack of assurance of good strength retention.

(d) Large parts are not required in sufficient volume to warrant cost of matched tools.

(e) Parts are under development or subject to change. (Internal changes can be made in many cases without affecting tools at all. Mold shape changes are accomplished on female mold only.)

It also provides for lapping or joining of cut glass patterns without fear of compression in a fixed gap resulting in resin starved areas and poor surface and strength properties.

Simpler, lower cost capital equipment is required. For example, one of our boats is molded using an internal pressure equivalent to a total load of approximately 865,000 lb. To obtain this pressure in a hydraulic press would require very elaborate and expensive equipment.

Actually, the two methods are not necessarily competitive, but supplementary. Each method has its advantages, each its limitations. If it is more economical to mold large parts to eliminate assembly or if structural advantage can be obtained by molding one large part instead of several smaller parts, then perhaps the bag method is the only way.

We utilize matched metal molds whenever we can. That means meeting all the above tests. If the molding can't be done feasibly with matched metal dies, then we use the bag or some other method. As a matter of fact, we consider both processes of equal importance to us and have development programs in progress in our plant and basic research programs in outside research laboratories.

Materials and Process Costs Must Be Brought Down

— George K. Hammond
Owens-Corning Fiberglas Corp.

OVER the last five years the average selling price of plastic reinforcements has reduced from roughly 50¢ to approximately 42¢ per lb.

There are several factors that will affect future prices very materially. The most important is the quantity of

material produced and used. A second is standardization or, in other words, fewer types and forms of fibrous glass and resin. Still another is the cost of resin raw materials, which can be reduced either through greater use of present polyester materials or through the use of less expensive materials of a new type.

We believe the price of fibrous glass, possibly in a different form than that used today, will go to 30¢ per lb in the next 5-year period, and can go to 20¢ or 25¢ per lb in the next 10 years. Polyester resins, on the other hand, probably cannot go below 35¢ per lb in the next five years unless new ingredients, fillers, or entirely new types based on other raw materials are developed. Resins manufacturers are constantly searching for lower costs without, and this is important, sacrificing quality.

There is certainly a large factor of cost in the process. In each dollar of selling price out of the molders' plants there is roughly only 16¢ worth of glass. If we halved our price, the per lb would still be 92¢. All three ingredients, the resin, the glass, and the process must cost less if we are to extend our economically competitive position with respect to steel.

There are at least four approaches to this problem. A combination of all in one process would materially reduce cost.

The first is to accomplish more in the molding stage. Build in color so only a final flash coat of paint is needed to finish the body, mold in all holes to eliminate after operations, and reduce the number of parts to a rock bottom minimum, possibly as few as 20 for a full-sized car, thus reducing assembly cost.

A second approach is to increase the glass loading through the use of cross-laid parallel strands. This would double the rigidity and permit sections possibly 0.070 in. thick instead of 0.100 or 0.125 in. A 30 to 40% cost reduction would result without sacrificing properties.

The third approach involves the use of strands and mats fully loaded with resin to reduce the molding cycle to seconds instead of minutes. The resin loaded strands could be cross-laid or chopped and preformed.

A final possibility is the use of high-strength molding compounds where an automatically weighed part could be dropped into the die, then flowed under 500 to 1000 psi to fill out the mold. Ribs and inserts can be readily included in such a process.

If we can combine these process improvements and material price reductions, the cost of fiber glass reinforced plastics should be reduced from the roughly $1 per lb of today to approximately 70¢ per lb. Adding the weight reduction accomplished through higher glass loadings, the probable figure comes to 45-50¢ per lb, or one-half of today's cost.

Design Limitations of Molded Fiber Glass Parts

– Robert S. Morrison
Molded Fiber Glass Body Co.

FIBER glass reinforcing mat puts the strength into an otherwise weak plastic. It is full of air. Now, we use polyester resin, a thermosetting liquid that turns solid and hard after being catalyzed and heated. This resin ties the glass fibers together, and gives body and rigidity to the combination. Polyester resin is used because it is a liquid and can be forced through the fiber glass mat without disturbing the position of the fibers, and it cures solid without creating any volatile gases. However, once cured it cannot be softened again by heat and formed a second time.

The main problem is to force all of the air out of the fiber glass mat with the resin so there will be no pinholes in it, and so that it can be painted easily.

Another problem is to be sure the glass fibers are evenly distributed throughout the molded part. I mentioned before that the plastic alone is weak, so any spot without the fiber glass would be weak. Also, if we get too much fiber glass in any one spot the resin cannot penetrate it and a dry spot will result.

Polyester resin has no resistance to flow, while most other plastic materials resist flow so much that pressures of 2000 psi are needed. We can flow polyester resins with very little pressure, which is a great advantage and a disadvantage. With no resistance to flow, the only way we can put pressure on the resin is to trap it in the dies and not let it escape. We can say that we put hydraulic pressure on the resin, using the same principle as in hydraulic brakes.

So, our dies must telescope into each other, like a piston into a cylinder, and the resin must be trapped in between the dies before the telescoping action is completed. We call the sections of the two dies that telescope together and trap the resin, as well as shearing off the glass and making the part to exact size, the pinch-offs, which are flame-hardened and are fitted to have a clearance of 0.002 in. at all points.

In addition to the above points, there are two other basic factors that limit the design of molded fiber glass parts. First is the bulk factor of the fiber glass mat itself, which is about five times as thick as the final molded part is. On parts with very steep sides the material tends to wrinkle as one die slides into the other die. Second, polyester resins shrink in curing, so that draft is needed on all parts to prevent locking on the male die.

These general characteristics listed above create the following limitations on molded fiber glass parts design:

1. All parts must have draft, and no undercuts or returns can be molded.
2. Because of the difficulty in locating the fiber glass reinforcement accurately, parts with different wall thicknesses should be avoided wherever possible.
3. No postforming can be done after molding.
4. Sharp corners of any type should be avoided to avoid resin-rich weak corners.
5. Vertical pinch-offs do not hold resin properly and also drag the glass fibers out of place during the telescoping of the dies. Parting lines should be as horizontal as possible. Frequently, to avoid vertical pinch-offs, we mold a part double and cut it in half, or even mold an extra section, cut it off, and throw it away.
6. Due to resin shrinkage, die cleaning problems, and the like, inside holes should not be molded unless punching after molding is difficult or impossible. Inside holes should be designed so vertical punching is practical.
7. The edge of the molded piece will be either horizontal or vertical to the die plane because of telescoping pinch-off. This means that a contoured part, such as a rear deck, may not have the angle of its edge surface in proper car position, which may require some hand grinding after molding. This might be especially true if an inner and outer deck are matched together.
8. Changes in direction of flanges create a condition where the telescoping pinch-off must be reversed. At this point of reversing the pinch-off, it is very difficult to attain proper sealing and fitting of the dies. Reverse pinch-offs should be avoided wherever possible.
9. Another limitation on the use of matched metal dies is, of course, the time to get them produced and into production. This time will run anywhere from four weeks to eight months, depending upon the size and complexity of the die and the availability of the particular machine tools needed to produce the dies.

I have mentioned these design limitations not to discourage the use of molded fiber glass automotive parts but to help people involved in the design of such parts avoid designs that are difficult or impossible to mold. This will actually result in lower cost and higher production rates and higher quality parts, all of which are necessary if molded fiber glass car bodies are to be produced in large volume.

Scavenging

THE indicated output of a 2-stroke engine is primarily dependent upon the success with which the products of combustion are driven from the cylinder and are replaced by fresh air or mixture during the scavenging period. Such replacement must, of course, be accomplished with a minimum of blower power.

This paper deals with various aspects of 2-stroke research conducted at M.I.T. during the past 10 years. Among the subjects discussed are the methods used in the prediction and measurement of scavenging efficiency, and the effect of engine design and operating variables on the scavenging blower requirements as reflected by the scavenging ratio.

THE staff of the Sloan Laboratory at Massachusetts Institute of Technology has always been interested in the 2-stroke cycle, and in engine developments in this field. Between 10 and 15 years ago, under the sponsorship of the National Advisory Committee for Aeronautics, a 2-stroke cylinder was designed and built for research on port shape and timing. Curtis-type porting was used and the cylinder was split longitudinally to permit the inlet or exhaust ports to be moved separately up or down to vary the timing. The ports were extra long so that the vertical height and angle of each port could be changed by installing suitable inserts. Much useful information was obtained with this cylinder, which has already been reported in the literature,[1, 2, 3] so that it will not be covered here. This engine is still proving of value, and it is urged that those interested in loop-scavenged engines build a similar one for studies of porting, timing, scavenging, and control.

In the 2-stroke engine, a more or less separate mechanism known as the scavenging pump supplies air or fresh mixture to the inlet ports at a pressure higher than exhaust-system pressure. The process by which this fresh air drives the combustion products from the cylinder, and is itself trapped in the cylinder, is called scavenging. Since the ability of the engine to produce power depends upon the amount of fresh air that can be trapped in the cylinders per unit time, one of the principal objectives of the designer of this type of engine is to learn how to trap reasonable quantities of fresh air without the expenditure of undesirable amounts of scavenging-blower power.

Granting that scavenging is important, it is desirable that the designer not only have an ideal of scavenging excellence at which to aim, but that he should also be able to determine without too much difficulty how closely his engine approaches this ideal. During the last few years our laboratory has gradually settled upon terminology and methods by means of which actual scavenging may be compared with an ideal scavenging process. The results of this work have been used to re-examine, to a limited extent, the effect of engine design on scavenging. A separate investigation of the effect of operating variables on scavenging pressure requirements has also been made.

Fig. 1 shows several types of 2-stroke cylinders. The principles on which they operate are the same in spite of the differences in mechanical design and arrangement. These engines all have *ports* which are uncovered by the piston near bottom center. Cylinders with inlet and exhaust ports at one end, such as types (a) and (b), are often called *loop* scavenged, while cylinders with inlet ports at one end and exhaust ports at the other (types c through f) are said to be *through* scavenged. For convenience, these terms will be used here.

Ideal Scavenging – In a perfect scavenging process the fresh mixture is considered to push the combustion products before it without mixing or exchanging heat. This process continues until all the burned gases have been expelled and the cyl-

[1] See NACA TN 674 (1938), "Scavenging a Piston-Ported 2-Stroke Cylinder," by A. R. Rogowski and C. L. Bouchard.
[2] See NACA TN 756 (1940), "Effect of Piston-Head Shape, Cylinder-Head Shape, and Exhaust Restriction on Performance of Piston-Ported 2-Stroke Cylinder," by A. R. Rogowski, C. L. Bouchard, and C. F. Taylor.
[3] See NACA TN 919 (1943), "Part-Throttle Operation of Piston-Ported 2-Stroke Cylinder," by A. R. Rogowski and C. F. Taylor.

the 2-Stroke Engine

C. F. Taylor, and A. R. Rogowski,
Massachusetts Institute of Technology

This paper was presented at the SAE National Diesel-Engine Meeting, Chicago, Nov. 3, 1953

inder is completely filled with fresh mixture. The temperature of the mixture is assumed, for simplicity, to be the same as that in the inlet system. In loop-scavenged engines, since the exhaust ports remain open after the inlet ports close, there is a strong tendency for the cylinder pressure to drop close to exhaust pressure before exhaust ports close, as shown in Fig. 2. Even with through-scavenged engines, where the exhaust port closes slightly before the inlet, it is found that the pressure during the compression stroke is seldom greater than it would be if compression had started at bottom center with exhaust-system pressure in the cylinder. In view of this practical limitation, our ideal for scavenging is to retain within the cylinder a mass of fresh mixture equal to that which would just fill the entire cylinder volume at inlet temperature and exhaust pressure.

It will be noted that, in the idealized scavenging process above described, not only is the cylinder completely filled with fresh mixture but also no fresh mixture escapes from the exhaust ports, and thus all of the mixture supplied remains to take part in the subsequent combustion and expansion.

Nomenclature – Under the ideal conditions specified above, the mass of dry air present in the cylinder after scavenging will be:

$$M_r^* = \rho_s V_c \qquad (1)$$

where ρ_s is the density of *dry air* in the inlet mixture based upon inlet temperature and exhaust pressure. If dry air is being used for scavenging, then from the gas law:

$$\rho_s = \frac{p_e}{53.3\ T_i}$$

where p_e is the absolute exhaust pressure and T_i is the absolute inlet temperature. If water vapor or gaseous fuel is present in the scavenging mixture, ρ_s must be corrected to compensate for the partial pressures of these components.

$$V_c = V_{disp} \times \left(\frac{r}{r-1}\right)$$

where V_c is the total cylinder volume and r is the compression ratio.

The mass of dry air actually delivered through

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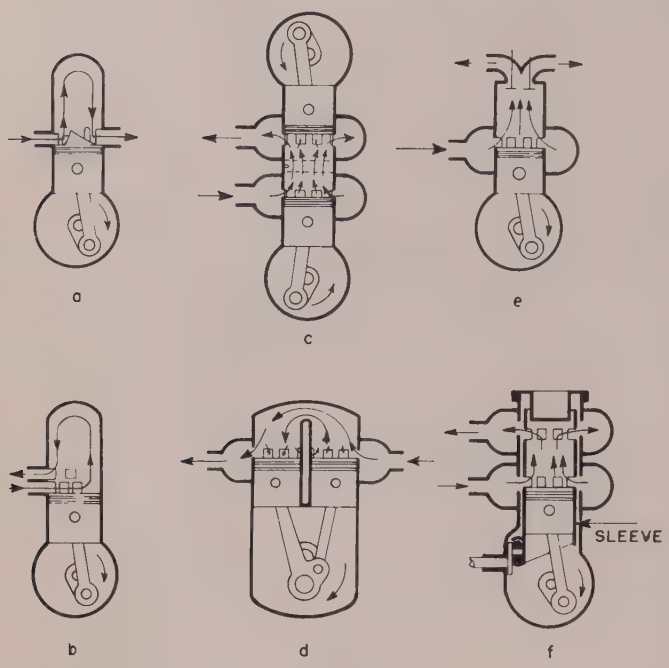


Fig. 1 – 2- stroke engine types: a – loop scavenged, b – loop scavenged, c – opposed piston, d – U cylinder, e – poppet valve, f – sleeve valve

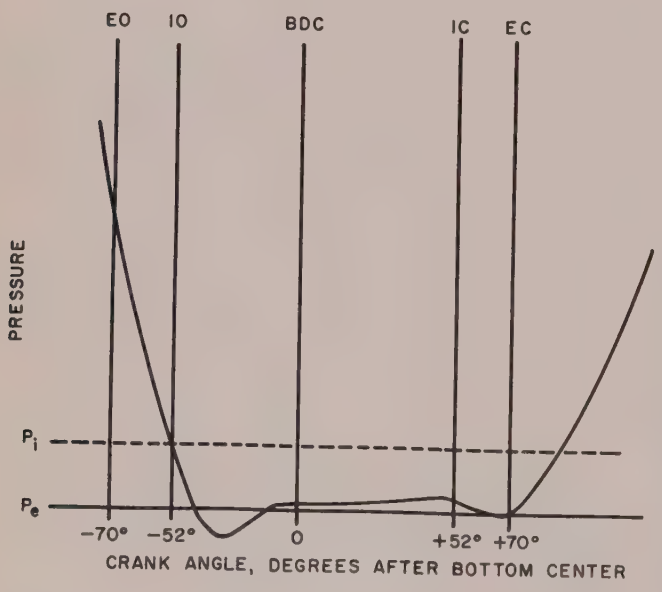


Fig. 2 – Diagram for pressure versus crank angle – engine of type a (Fig. 1) – M.I.T. cylinder, 4½ x 6, 1800 rpm,

$R = 7$, $R_s = 1.4$, $p_e/p_i = 0.74$, $P_s = 14.7$ psia

(from Fig. 26 of Rogowski and Bouchard[1])

the inlet ports of the cylinder per cycle, divided by the above ideal air quantity, is called the *scavenging ratio*, R_s.

Thus:

$$R_s = \frac{M/N}{\rho_s V_c} \qquad (2)$$

where M is the mass flow of dry air delivered to the cylinder per unit of time, and N is the number of cycles per unit of time. For cylinders scavenged with dry air only:

$$R_s = \frac{M}{N V_c} \times \frac{53.3 T_i}{p_e}$$

The mass of *fresh* dry air *retained* in the cylinder after the ports close, divided by the ideal air quantity, is known as the *scavenging efficiency*, e_s.

Thus:

$$e_s = \frac{M_r/N}{\rho_s V_c} \qquad (3)$$

where M_r is the mass of fresh dry air retained in the cylinder per unit time.

A third useful quantity is the *charging efficiency*, Γ', the fraction of the fresh air supplied that is actually retained. From the above definitions it is evident that:

$$\Gamma = \frac{e_s}{R_s} = \frac{M_r}{M}$$

From these definitions it also follows that:

$$\text{Imep} = e_s \rho_s \left(\frac{r}{r-1}\right) (F' Q_c \eta_i') \frac{778}{144} \qquad (4)$$

where:

Imep = Indicated mean effective pressure, psi

F' = [Cylinder fuel/air ratio = $\dfrac{\text{fuel retained}}{\text{fresh air retained}}$

η_i' = Cylinder thermal efficiency, based on fuel retained

Q_c = Heating value of the fuel, Btu per lb

It is evident that for a carbureted engine F' is the same as the fuel/air ratio supplied by the carburetor, while the actual indicated thermal efficiency $\eta_i = \Gamma \eta_i'$, due to loss of fuel through the exhaust ports during scavenging.

For engines in which fuel is injected after ports are closed, since no fuel is lost $\eta_i = \eta_i'$.

For purposes of estimation, η_i' may be taken as 80% of the fuel-air cycle efficiency, for the same F' and compression ratio.

The fuel-air cycle is a theoretical engine cycle in which cognizance is taken of the actual thermodynamic and chemical characteristics of the cylinder charge.[4, 5] Figs. 3 and 4 show curves of

[4] See "Internal-Combustion Engine," by C. F. Taylor and E. S. Taylor. Pub. by International Textbook Co., Scranton, Pa., 1948.

[5] See SAE Transactions, Vol. 31, (October) 1936, pp. 409-424: "Thermodynamic Properties of Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel.

fuel-air cycle efficiency versus fuel/air ratio and compression ratio, for constant-volume and limited-pressure cycles. Under comparable conditions (same compression ratio, same fuel/air ratio, and same pressure limit) the ratio between actual indicated thermal efficiency and the efficiency of the fuel-air cycle is almost constant for a given type of engine, over most of the useful operating range.

Relationship between Scavenging Ratio and Scavenging Efficiency – In actual engines, some of the fresh mixture acts as in perfect scavenging. On the other hand, some mixes with combustion products and is forced out with them and is lost, and some of the fresh mixture may "short circuit" across the cylinder and flow out the exhaust ports without producing any scavenging effect at all. In actual engines, the exact relationship between scavenging ratio and scavenging efficiency can be determined only by tests, but some idea of the possibilities may be gained from an examination of Fig. 5. Curve (a) represents perfect scavenging in which no mixing or heat exchange takes place between fresh mixture and combustion products, and no fresh mixture is lost until the cylinder is filled at scavenging density. Therefore, at scavenging ratios between zero and unity, all mixture supplied is also retained, scavenging efficiency is equal to scavenging ratio, and the charging efficiency is unity. At higher scavenging ratios the scavenging efficiency does not increase, as the cylinder is already completely filled with fresh mixture.

Curve (b) is based on the assumptions that (1) the fresh mixture mixes *completely* and *instantaneously* with the residual gases as it enters the cylinder and (2) the residual gases are at the same temperature and have the same molecular weight as the fresh mixture. As a given mass of fresh mixture enters the cylinder an equal mass of exhaust gas and fresh mixture escapes through the exhaust ports.

An analysis based on these assumptions leads to

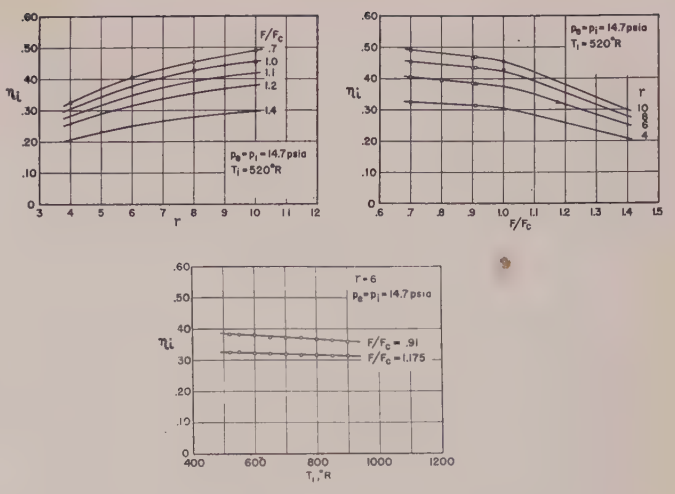


Fig. 3 – Indicated efficiencies of constant-volume fuel-air cycle

η_i = indicated thermal efficiency, p_e = absolute exhaust-system pressure, p_i = absolute inlet pressure at inlet port, T_i = absolute temperature at inlet port, r = compression ratio, F = fuel/air ratio, F_c = chemically correct fuel/air ratio, fuel was C_8H_{18} (computed from charts of Hershey et al[5])

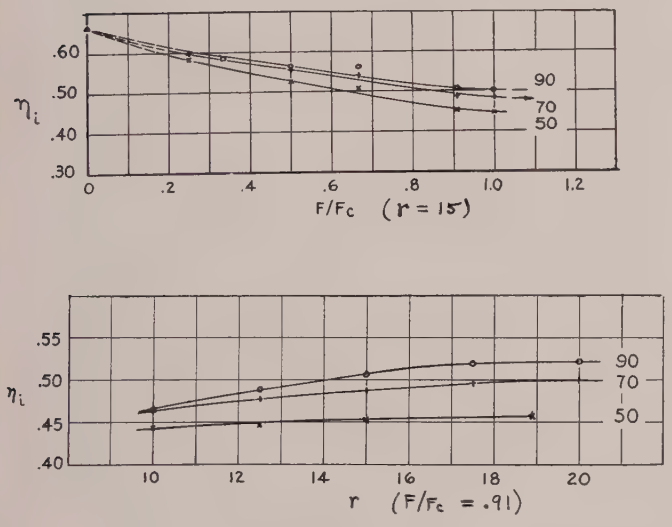


Fig. 4 – Indicated efficiencies of limited-pressure fuel-air cycle (same symbols as in Fig. 3)

T_i = 520 R, $P_i = P_e$ = 14.7 psia – numbers on curves indicate ratio of maximum pressure to inlet pressure (computed from charts of Hershey et al[5])

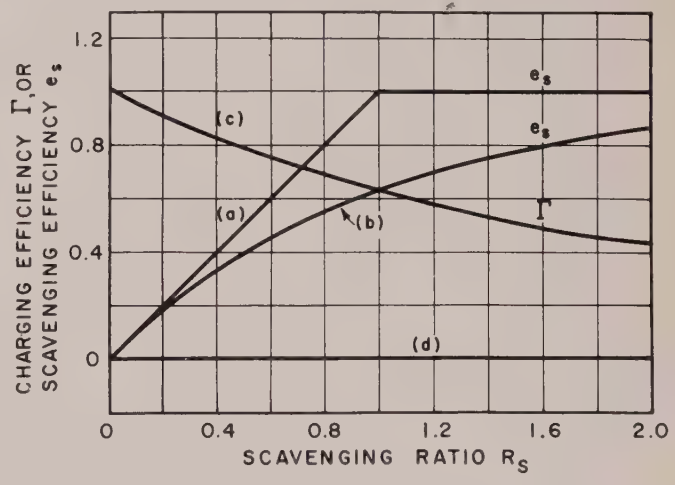


Fig. 5 – Theoretical relationships between scavenging efficiency, charging efficiency, and scavenging ratio

(a) e_s with perfect scavenging; (b) e_s with perfect mixing, $e_s = 1 - \epsilon^{-R_s}$; (c) Γ with perfect mixing; (d) e_s with complete short circuiting

the theoretical relationship:

$$e_s = 1 - \epsilon^{-R_s} \qquad (5)$$

where ϵ is the base of the natural logarithms.

Curve (b) is a plot of this relationship, while curve (c) gives the corresponding values of Γ.

Curve (d) represents complete *short circuiting.* In such a case, the scavenging efficiency would remain near zero at any value of the scavenging ratio.

The curve of e_s versus R_s for a given engine will be some composite of curves (a), (b), and (d). In practice it is found that most engines show a characteristic curve close to curve (b), although this coincidence does not mean that the actual process is necessarily one of complete mixing.

Measurement of Scavenging Efficiency – The evaluation of a given 2-stroke engine design is incomplete and the possibilities for future improvement in power output are unknown unless its actual curve of scavenging efficiency versus scavenging ratio is measured.

Measurement of *scavenging ratio,* or *air supplied,* is relatively simple. We usually use an ASME standard square-edge orifice with flange taps.[6]

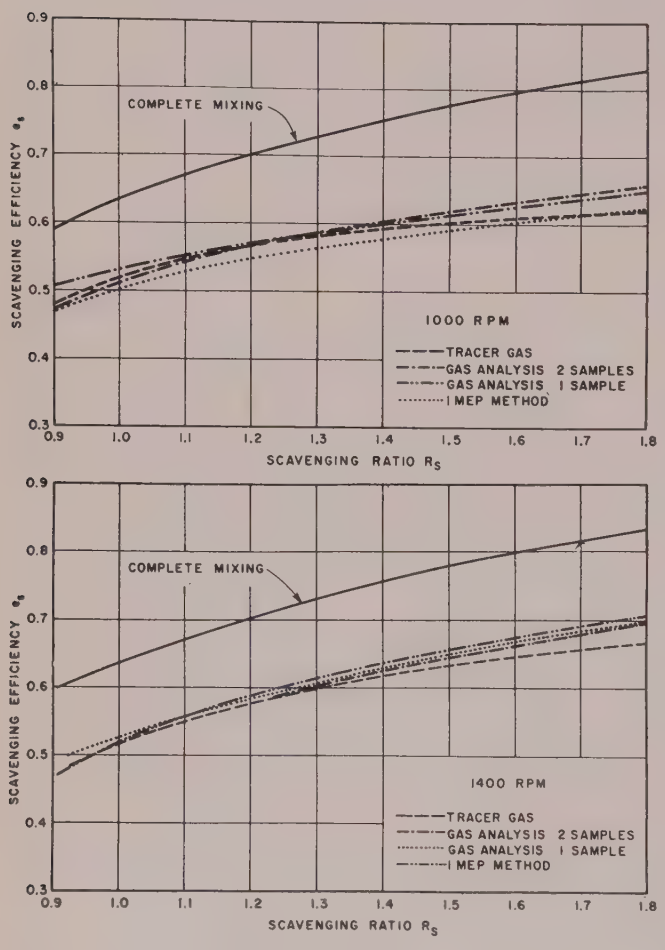


Fig. 6 – Scavenging efficiency as determined by four methods – M.I.T. 3¼ x 4½ in. cylinder (see footnote 7)

Suitable surge chambers must be used between orifice and engine to eliminate pulsations at the orifice. For accurate measurement of inlet temperature, the thermometer must be placed where it will not be influenced by radiation or by exhaust gas blowing back through the inlet ports. Accurate measurement of inlet and exhaust pressures may be assured by making the measurement in large inlet and exhaust receivers.

Measurement of *scavenging efficiency,* or *fresh air retained,* is difficult, as there is no direct method of determining what portion of the fresh mixture remains in the cylinder after scavenging. We have recently compared, in the laboratory, four methods of determining scavenging efficiency. The graduate research team carrying out this work consisted of Officers of the U. S. Navy.[7]

The engine used for this comparison was a 3¼ x 4½ in. loop-scavenged single-cylinder unit, mounted on a standard CFR crankcase, and operated as a spark-ignition engine with cylinder injection of the fuel. Best-power fuel/air ratio was used, and best-power spark advance. Scavenging ratio was varied from 1.0 to 1.8 at two speeds, 1000 and 1400 rpm (750 and 1050 fpm mean piston speed).

Tracer Gas Method – This method, described by Schweitzer and DeLuca,[8] is based on the use of small quantities of a gas that will be chemically destroyed at combustion temperatures, but will not be changed at temperatures below those prevailing in the cylinder when the exhaust ports open. If such a gas is thoroughly mixed with the fresh mixture, then:

$$e_s = \Gamma R_s = \frac{x - y}{x} R_s \qquad (6)$$

where x is the mass flow of tracer gas into the engine and y is the mass flow of unburned tracer gas out of the exhaust. x can be measured by measuring the inflow of tracer gas, y can be obtained by chemical analysis of the exhaust gases and measurement of the mass flow of air and fuel through the engine. In actual use it is necessary to correct the above expression by assuming that some small fraction of the tracer gas retained in the cylinder will escape dissociation. Some assumption must also be made regarding the dissociation or absorption of the tracer in the exhaust system before sampling takes place.

The tracer used in our tests was monomethylamine. In attempting to sample downstream of the exhaust surge tank to assure good mixing and a representative sample, we found that the tracer was readily absorbed by condensate in the surge

[6] See "Fluid Meters, Their Theory and Application – Part I." Pub. by ASME, New York, 1937.

[7] "Measurement of Scavenging Efficiency of 2-Stroke Engine – Comparison and Analysis of Methods," by A. L. Waldron, et al. M.I.T. thesis, 1949.

[8] See NACA TN 838 (1942), "Tracer Gas Method of Determining Charging Efficiency of 2-Stroke-Cycle Diesel Engines," by P. H. Schweitzer and F. DeLuca.

tank. With the high exhaust temperatures produced by the spark-ignition engine, great difficulty was experienced in avoiding dissociation in the exhaust stream. It was found necessary to make multiple openings in the exhaust sampler, to install it close to the exhaust ports, and to sample at high velocity in order to remove the tracer quickly from the high-temperature region. Only after these steps were taken could consistent and reasonable results be obtained. It was estimated that 20 man-hr were required to set up the apparatus and about 25 min to obtain a sample and determine one point on the e_s versus R_s curve. Using all these precautions, we obtained curves (Fig. 6) that are in good agreement with those obtained by other methods. However, in view of the likelihood of serious error when the method is improperly used and because of the time required, it is felt that the tracer gas method requires further development.

Gas Analysis of Compression and Expansion Samples – In this method a Cox indicator unit was used as a sampling valve. A gas sample was taken just before exhaust-port opening to establish by analysis the fuel/air ratio and per cent of oxygen present. Since some of this same gas remains in the cylinder as residual, the per cent oxygen in the residual will thus be determined. A sample is also taken about 60 deg after port closing on the compression stroke to determine the per cent oxygen in the mixture of fresh charge and residuals in the cylinder. As the per cent of oxygen in the entering fresh mixture is known, an equation involving the oxygen balance may be solved for scavenging efficiency. Results using this method were quite consistent. The setup is simple, provided an opening in the cylinder head is available for sampling.

This method involves two samples for each run. As in any sampling method, it is necessary that the samples be representative of the cylinder contents. In addition to the samples, an estimate of the temperature of the residual gases at the time of mixing with the fresh charge must be made, using fuel-air cycle and heat-transfer calculations. Because of these difficulties and the time required to obtain one point on the scavenging curve (1½ hr), this method was considered the least desirable of those tried.

Indicated Mean Effective Pressure Method – This method applies only to spark-ignition engines scavenged with a homogeneous gaseous fuel-air mixture of known composition. It has been used quite successfully at M.I.T. It is considered the only reliable and practical method for this type of engine, but it is a laboratory method and requires the use of considerable equipment.

This method depends on the relationships expressed in equation (4). For a given engine test, ρ_s and Q_c are known, and imep is measured from an indicator card or from brake hp and friction hp tests. F' is the fuel/air ratio within the cylinder during combustion, which, in engines of this type is equal to F, the fuel/air ratio supplied by the carburetor. The remaining unknowns are e_s and η_i'.

The most reliable method of estimating η_i' is to operate a 4-stroke engine of nearly the same cylinder size, with the same compression ratio and the same fuel-air mixture, and to measure its indicated thermal efficiency based on the compression and expansion strokes only. It is then assumed that η_i' for the 2-stroke engine will be equal to x times the indicated efficiency of the 4-stroke engine, where x is the correction factor accounting for the smaller area of the "toe" of the 2-stroke diagram. This characteristic shape, shown in Fig. 7, is due to the fact that 2-stroke exhaust ports must open well before bottom center in order to reduce blowback into the inlet system. The value of x may be measured by comparing the 2-stroke and 4-stroke indicator diagrams. It usually lies between 0.92 and 0.97, and can safely be taken as 0.94 when measurements are not available.

In order that η_i' of the 2-stroke engine equals that for the 4-stroke engine (except for the correction factor suggested above), in addition to having the same compression ratio and fuel/air ratio, the following conditions must be fulfilled:

(a) The fuel/air ratio used should be near that for maximum imep, both to reduce errors in measurement of imep and to reduce the influence of differences in residual-gas content on efficiency.

(b) The 4-stroke engine should not have a large valve-overlap angle, since escape of fresh mixture

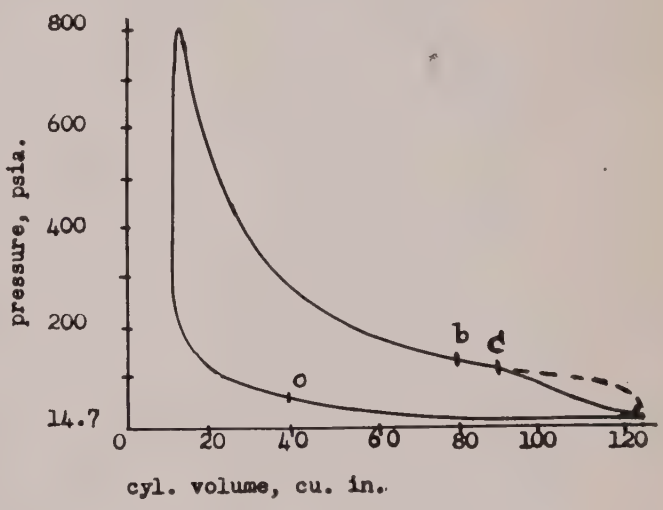


Fig. 7 – Indicator diagram from 2-stroke engine, opposed-piston type. Bore 3¾ in., stroke 5 in. for each piston, compression ratio 10. Fuel was injected after port closing. Exhaust valve starts to open at c. Dash line shows diagram for corresponding 4-stroke engine. Imep = 160 psi (4-stroke imep = 170 psi)

during the overlap period would affect the efficiency of the 4-stroke engine to an unknown extent.

(c) Best-power spark advance should be used in all cases.

(d) The two engines should have nearly the same cylinder dimensions and should be run at the same piston speed and jacket temperatures, so that losses during combustion and expansion will be nearly the same.

In the absence of a suitable comparative test on a 4-stroke engine, it may be assumed with reasonable assurance that η_i' for a 2-stroke engine scavenged with a homogeneous gaseous fuel-air mixture will be 0.80 times the efficiency of the corresponding fuel-air cycle (Figs. 3 and 4), provided optimum spark timing is used and the fuel/air ratio is near that for maximum imep.

In our work with this method, x and η_i' were measured by means of indicator diagrams taken with the M.I.T. point-by-point indicator on both the 2-stroke cylinder and a 4-stroke (CFR) cylinder of the same size.

After these values had been established, we again used the engine indicator to determine the imep of the 2-stroke engine for each point on the e_s versus R_s curve. The time required to establish a point in this manner was 1¼ hr. In most cases it will be sufficiently accurate to determine imep from brake hp and motoring friction hp. When this method is used, the determination of a point on the e_s versus R_s curve should take only 5 or 10 min. No difficulties with technique or equipment were experienced with this method.

Gas Analysis of Expansion Sample – This method applies to diesel engines, and to spark-ignition engines in which fuel is injected after the ports close. The method is fairly simple, gives consistent results, and may be used in the field with relatively simple equipment. It is the only method that we can recommend for this type of engine. In the use of this method, a gas sample is taken near the end of the expansion stroke. The sample is analyzed for CO_2, O_2, and CO. From curves of exhaust-gas composition versus fuel/air ratio obtained from 4-stroke engines with small valve overlap, such as those of Fig. 8 and several references,[9, 10, 11] the fuel/air ratio that must have existed within the cylinder $\left(\text{that is, } F' \text{ or } \frac{\text{lb of fuel retained}}{\text{lb of air retained}}\right)$ is determined.

With three exhaust constituents it is possible to obtain a good check on F'. Since fuel is injected after port closing, fuel injected = fuel retained. Fuel injected is easily measured, so that:

$$M_r = \frac{\text{Fuel injected per unit time}}{F'}$$

where M_r is air retained per unit time.

It may be of value to point out here that the presence of residual gas left in the cylinder need not be considered in this method, since the composition of the residual gas is exactly the same as that of the products of combustion that are sampled. The composition of the sampled gas can thus be considered to be the result of the combustion of the fresh fuel and fresh air retained in the cyl-

[9] See SAE Transactions, Vol. 31, (March) 1936, pp. 90-98: "Relation of Exhaust-Gas Composition to Air/Fuel Ratio," by B. A. D'Alleva and W. G. Lovell.

[10] See NACA TR 757 (1943), "Relation of Constituents in Normal Exhaust Gas to Fuel/Air Ratio," by H. C. Gerrish and J. L. Meem.

[11] See SAE *Quarterly* Transactions, Vol. 4, July, 1950, pp. 330-346: "Composition of Diesel Exhaust Gas," by M. A. Elliott and R. F. Davis.

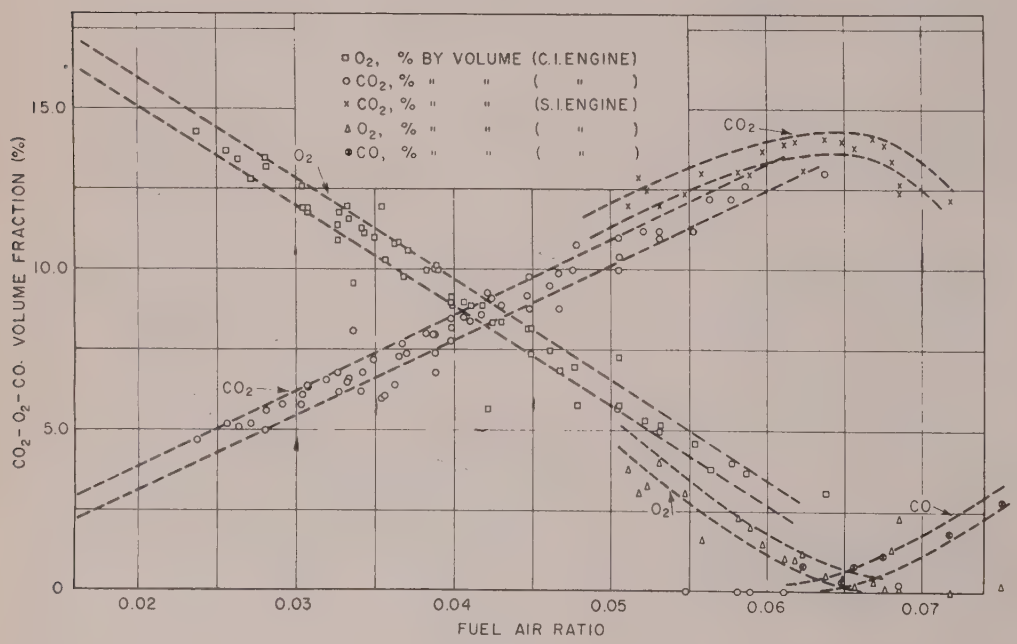


Fig. 8 – Composition of exhaust gases from 4-stroke spark-ignition and compression-ignition engines (see footnote 12)

under. If the fresh fuel happens to take oxygen from the residual gas it leaves that much more oxygen in the fresh air, so that the total per cent of oxygen present in the cylinder will be the same.

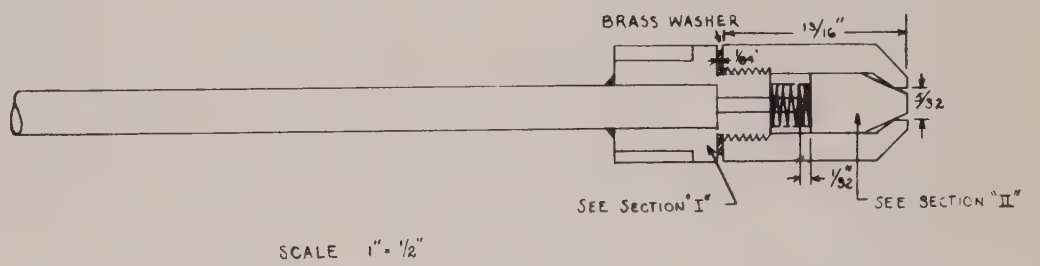
The above method gave the most reproducible results of those tried. Sources of error include the normal errors in gas analysis and the possibilities that the sampling valve does not draw a representative sample or that the engine does not follow the published curves of exhaust composition. Setup time for this method is about one man-hour. Time to obtain one point on the e_s versus R_s curve is about 25 min.

Comparison of Methods of Measuring Scavenging Efficiency – Fig. 6 shows that when great care has been used in minimizing sources of error, all of the foregoing methods give substantially the same result. It should perhaps be re-emphasized, however, that great differences exist between methods with regard to the amount of technique and time required to achieve reproducible results.

Our experience has led us to the conclusion that, for engines scavenged with a carbureted mixture, the imep method is to be preferred, provided imep can be measured either by a balanced-pressure type of indicator or by carefully made firing and motoring tests. The tracer gas method is not suited to this type of engine on account of its high exhaust temperature, and the 2-sample method is both cumbersome and unreliable. For engines in which the fuel is injected after port closing, use of a single expansion sample is both the most convenient method and the one least likely to involve serious error.

Impact Sampler – The usefulness of the last mentioned method of determining scavenging efficiency has been greatly improved by the development of a semiautomatic impact check-valve sampler, for use in place of the timed sampling valve. The impact sampler is shown in Fig. 9. It has already been described by Houtsma et al.[12] This instrument consists of a small tube, the open end of which carries a check valve opening inward. The tube is placed in the exhaust port in such a way as to face into the exhaust-gas stream at the point where it first emerges from the cylinder. The tube opening is thus exposed to a total pressure approximately equal to the cylinder pressure during the early part of the exhaust process. The tube is connected to a small receiving tank with a bleed valve to atmosphere. This valve is adjusted so as to hold the pressure in the tank above the scavenging pressure. Thus, the check valve closes before the cylinder pressure falls to scavenging pressure, and no fresh gases enter the sampling tube during the scavenging process. In taking a

[12] "Correlation of Scavenging Ratio and Scavenging Efficiency in 2-Stroke Compression-Ignition Engine," by C. G. Houtsma, et al. M.I.T. thesis, 1950.



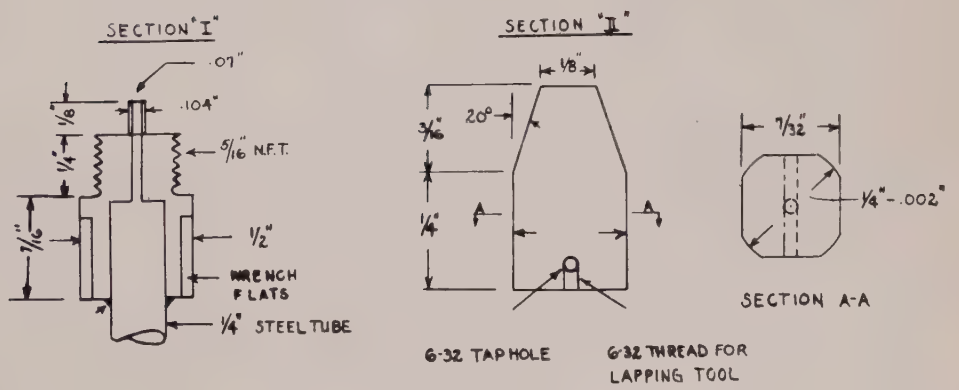


Fig. 9 – Details of sampling valve – all material is mild steel (see footnote 12)

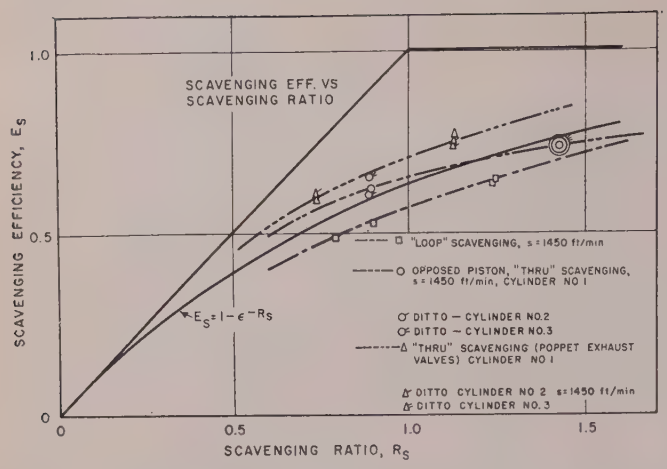

SCAVENGING EFF. VS SCAVENGING RATIO
SCAVENGING EFFICIENCY, E_S
SCAVENGING RATIO, R_S
$E_S = 1 - \epsilon^{-R_S}$
"LOOP" SCAVENGING, s = 1450 ft/min
OPPOSED PISTON, "THRU" SCAVENGING, s = 1450 ft/min, CYLINDER NO 1
DITTO — CYLINDER NO.2
DITTO — CYLINDER NO.3
"THRU" SCAVENGING (POPPET EXHAUST VALVES) CYLINDER NO 1
DITTO CYLINDER NO 2 s = 1450 ft/min
DITTO CYLINDER NO. 3

Fig. 10 – Scavenging characteristics of three engines as determined by impact sampler (see footnote 12)

sample, the gases are allowed to flow through the sample receiver long enough to make sure that it is well purged before the contents are analyzed. Analysis may be by the usual Orsat apparatus or its equivalent. In the process of development it was found that cooling was unnecessary, leading to the simple construction shown in Fig. 9. The impact area must be large enough compared with the mass of the valve to ensure quick operation. The great difference between the seat and valve cone angles results in line contact. This means that the area exposed to blowdown pressure can be considered equal to the area exposed to the sampling pressure. The travel of the valve is limited to about one-quarter of the diameter of the seat.

This sampling valve has the advantage that no extra hole is required in the cylinder. In practice, sampling may be done by introducing the valve through pyrometer openings, through special fittings in the exhaust manifold, or by direct installation in the port. After some experimentation with location it has always been possible to produce sampling pressures far above scavenging pressure. Timing is automatic and no timing mechanism is necessary. Since the sampling pressure is always kept higher than the scavenging pressure, the only effect of valve leakage is to reduce the maximum sampling pressure that may be maintained; accuracy of sampling is not affected. The weak spring shown is used to start the buildup of back pressure. When this buildup has once been accomplished, the spring is no longer necessary. One of the advantages of the check valve over the timed sampling valve in the cylinder head is its ability to extract exhaust-gas samples over the whole range of pressure from maximum blowdown pressure to 1 in. Hg or so above scavenging pressure. Thus, sampling of the exhaust gas takes place during most of the blowdown period, and a representative sample is assured. With a timed cylinder-head sampling valve, there is always danger that stratification in the cylinder will affect the results, particularly at light loads in diesel engines.

Effect of Engine Type on Scavenging Efficiency – Since few data are available on the scavenging efficiency of 2-stroke engines in general, or even of particular types, the improved impact type of sampling valve was used in a series of tests[12] on one loop-scavenged engine, and two through-scavenged engines; one an opposed-piston type, the other with exhaust valves in the cylinder head. (This work was done by a graduate research team consisting of Navy and Coast Guard officers.)

The impact sampler was first used in the exhaust ports of a 4-stroke spark-ignition engine and a 4-stroke diesel engine. F^1 may be determined for 4-stroke engines with small overlap, by direct measurement of fuel and air supplied to the engine. The curves of CO_2, O_2, and CO shown in Fig. 8 were thus obtained. These curves checked well

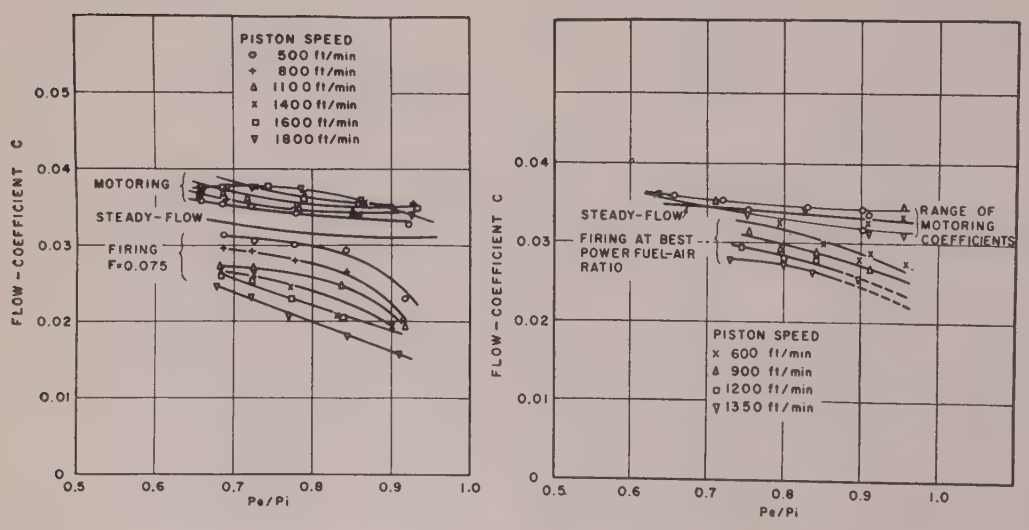


Fig. 11 – Effect of absolute exhaust pressure/absolute intake pressure and piston speed on orifice flow coefficient – left: 4½ x 6 cylinder; right: 3¼ x 4½ cylinder (see footnote 13)

with those of others.[9, 10, 11] Analyses of the gases sampled during blowdown were compared with these curves to determine the cylinder fuel/air ratio. Runs at low scavenging ratio were made by arbitrarily reducing the scavenging airflow and adjusting the load until the original cylinder fuel/air ratio was restored as indicated by the gas sample.

The results of this investigation are shown in Fig. 10. It is seen that the through-scavenged engine with valves in head has a scavenging efficiency about 20% higher than the loop-scavenged engine. The opposed-piston engine has a similar advantage at low scavenging ratios but is only about 10% better than the loop-scavenged engine at high scavenging ratio. The lower scavenging efficiency of the loop-scavenged engine is to be expected both because of the curved path that the scavenging air must take and because the port timing must be symmetrical. Such results depend heavily on detail design and should not be considered typical except in the most general way. In addition, the engines tested represented older models. Measurements of scavenging efficiency on many more engines will be necessary before the relative performance of these various types can be accurately evaluated.

Relationship between Scavenging Pressure and Scavenging Ratio – With a given cylinder and for a given scavenging ratio, the difference between inlet and exhaust pressure required increases very nearly as the square of the piston speed. Thus, with a given pump efficiency, the mean effective pressure required to drive the scavenging pump increases nearly in this ratio. It is thus evident that, when 2-stroke engines are called upon to operate at high piston speeds, the fraction of the engine power absorbed by the scavenging pump becomes very high, and it is not always the engine with the most favorable curve of scavenging efficiency versus scavenging ratio that has the greatest net output.

In order to study the pressure ratios across the engine required for scavenging, we have found it convenient to treat the cylinder as an orifice through which the scavenging air flows, due to the pressure difference between inlet and exhaust reservoirs.[13]

The mass flow of a gas through an orifice may be expressed as:

$$M = A\, C\, c_s\, \rho_i\, \phi \qquad (5)$$

where:

A = Orifice area, or a reference area proportional to orifice area

C = Orifice flow coefficient (for a fixed orifice C is nearly constant except at very low Reynolds numbers)

c_s = Speed of sound in the gas in the upstream reservoir

ρ_i = Density in the upstream reservoir

$$\phi = \sqrt{\frac{2}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{2/k} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]}$$

when p_2/p_1 is greater than 0.528

p_1 and p_2 = Pressures in reservoirs upstream and downstream of the orifice, respectively

As applied to an engine under static conditions, the value of C would depend upon the design of ports, piston, and cylinder, upon the crank position, and upon the reference area chosen. Piston area A_p is both convenient and significant as a reference area, and was used in our work.

Fig. 11 shows measurements of C made on two loop-scavenged single-cylinder engines.

Steady-flow coefficients were determined for various positions of the piston by measuring the flow of air through the engine, measuring reservoir pressures and temperatures, and solving for C at various ratios of p_e/p_i, (p_2/p_1). These values of flow coefficient were then averaged over the entire 360 deg of the cycle. The value of C for the engine so obtained may be thought of as the engine flow coefficient corresponding to zero piston speed, where the effects of gas-column inertia or blowback are absent. As shown in Fig. 11, it was found that the steady-flow coefficients vary only slightly with pressure ratio (p_e/p_i).

It was anticipated that the "chopping" effect, or intermittent stopping and starting of the flow in the inlet system, might cause C to be less under running conditions. To check this point, the engines were motored (without firing) over a range of piston speed, at various pressure ratios. Fig. 11 shows that the effect of p_e/p_i on C is of the same magnitude as in the steady-flow tests. It is interesting to see that the effect of piston speed on the motoring flow coefficient is small. This would seem to indicate that, for these two engines at least, the effect of intermittent flow in the inlet system need not be considered.

Fig. 12 shows values of C for the engines whose scavenging efficiency is given in Fig. 10. Figs. 11 and 12 both show that under firing conditions the flow coefficient is greatly reduced at high piston speed. The small effect of piston speed in the motoring runs and study of the indicator cards make it appear that the chief reason for the decrease in flow coefficient at high speeds is the larger crank angle occupied by exhaust blowdown. It was found that, at high piston speeds, even when reduced scavenging efficiency resulted in lower pressures in the cylinder at the instant of *exhaust* port opening, the cylinder pressures at *inlet* port opening were higher, resulting in greater blowback into the inlet system.

The firing flow coefficient is also strongly affected by p_e/p_i, being less at high values of p_e/p_i. This also

[13] "Investigation of Airflow in 2-Stroke Engines," by T. Y. Toong and D. H. Tsai. M.I.T. thesis, 1948.

appears to be chiefly the effect of blowback into the inlet system. Blowback depends on the relation between p_i and cylinder pressure at the time of inlet port opening. If p_i is increased, scavenging is improved and the cylinder pressure increases, but not usually as much as p_i itself. Thus, the blowback at low values of p_e/p_i will be less and the flow coefficient will be higher.

At very high values of p_e/p_i the flow coefficient is seen to rise again. The reason for this phenomenon may be that under these conditions scavenging is very poor, and the large amount of residual gases present in the cylinder will sharply reduce the cylinder pressure at exhaust and inlet opening, thereby reducing blowback. Another possibility is that under these conditions the exhaust dynamic effect may be large compared with the applied scavenging pressure difference.

Runs were made on one engine in which the fuel/air ratio was varied from 0.06 to 0.09. The effect of fuel/air ratio on flow coefficient or upon cylinder pressure at the time of inlet opening was found to be negligible. In diesel engines, where the range of useful fuel/air ratios is much greater, the effect of fuel/air ratio on flow coefficient may be important.

Relation of Port Flow Coefficient to Scavenging Ratio and Blower Power – It may be shown in terms of the above quantities that:

$$R_s = 2\,C\left(\frac{r-1}{r}\right)\left(\frac{c_s}{s}\right)\left(\frac{p_i}{p_e}\right)\phi \qquad (6)$$

where s is piston speed and the other symbols are as previously defined. This relationship shows that high values of C are desirable because, for a given inlet pressure, exhaust pressure, and piston speed, the scavenging ratio will be proportional to C. Conversely, as C decreases, higher inlet pressures, and therefore more scavenging blower power, will be required for a given value of R_s.

It may also be shown that the engine mep required to drive the scavenging blower can be obtained from the following equation:

$$Cmep = \frac{R_s\left(\frac{r}{r-1}\right)\rho_s\, c_p\, T_a\, y}{\eta_c} \times \frac{778}{144} \qquad (7)$$

where:

$Cmep$ = Engine mep required to drive the blower.

c_p = Specific heat of air at constant pressure (0.24 Btu per lb per deg F).

T_a = Compressor inlet temperature (usually atmospheric).

η_c = Compressor adiabatic efficiency.

$$y = \left[\left(\frac{p_i}{p_e}\right)^{\frac{k-1}{k}} - 1\right] \qquad (k = 1.4 \text{ for air})$$

The temperature rise of the air in the blower is $\frac{T_a y}{\eta_c}$, from which T_i and ρ_s may be calculated.

If the value of C for an engine is known or can be estimated, then R_s may be determined for any value of p_i/p_e, or vice versa, using expression (6). From a knowledge of C and the use of expression (7), therefore, we are able to calculate the engine power required to drive the scavenging blower for any value of scavenging ratio or piston speed.

Optimum Scavenging Ratio – With a given cylinder design operating at a given speed, power to scavenge can be reduced in two general ways, namely, (1) by reducing the scavenging ratio and (2) by increasing the port flow coefficient.

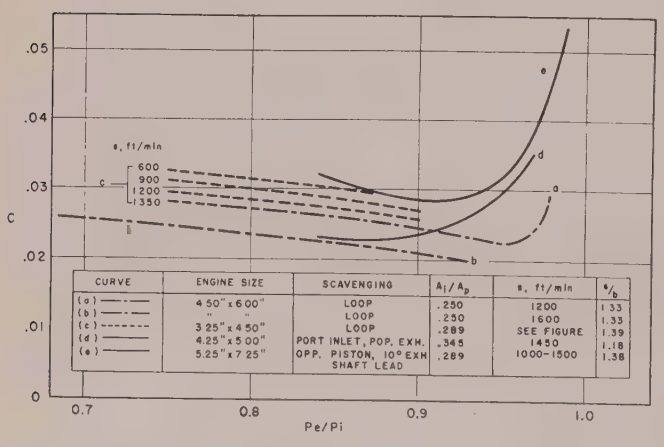
It is obvious that reducing the scavenging ratio will reduce the scavenging efficiency, and therefore reduce the imep. However, the net result may be favorable if the mep required to drive the scavenging pump is reduced even more.

Fig. 13 illustrates how, with a fixed port design, the optimum scavenging ratio becomes less as piston speed increases. These curves were computed for the M.I.T. loop-scavenged cylinder using measured scavenging efficiency versus scavenging ratio curves, and a port flow coefficient of 0.03. The optimum scavenging ratio obviously depends on the piston speed and on the relative importance of the maximum output and fuel economy.

Effect of Design on Flow Coefficient – The second possible method of reducing scavenging-pump power is by an increase in the flow coefficient. This coefficient is affected not only by the detail design of the ports but by the stroke/bore ratio and the type of cylinder used.

With piston-controlled ports, if the port height remains a given fraction of the stroke, the area available for ports is obviously proportional to bore x stroke, while the piston area is proportional to the bore squared. Thus, one useful way of achieving a high value of the port flow coefficient is the use of a large stroke/bore ratio. Designers should bear this in mind when laying out 2-stroke engines.

In both poppet-valve and opposed-piston engines,



CURVE	ENGINE SIZE	SCAVENGING	A_i/A_p	s, ft/min	$s/_b$
(a)	4.50" x 6.00"	LOOP	.250	1200	1.33
(b)	" "	LOOP	.250	1600	1.33
(c)	3.25" x 4.50"	LOOP	.289	SEE FIGURE	1.39
(d)	4.25" x 5.00"	PORT INLET, POP. EXH.	.345	1450	1.18
(e)	5.25" x 7.25"	OPP. PISTON, 10° EXH SHAFT LEAD	.289	1000–1500	1.38

Fig. 12 – Orifice flow coefficient versus absolute exhaust pressure/absolute intake pressure for various 2-stroke engines

a complete cylinder circumference is available for inlet porting. Thus, it is evident that, with ports of a given height in relation to the stroke and with equal stroke/bore ratio, the ratio of inlet port area to piston area will be twice as great in the poppet-valve cylinder as in the opposed-piston cylinder. In spite of this advantage, the poppet-valve cylinder of Fig. 12 has a lower flow coefficient than that of the opposed-piston cylinder, both because of its shorter stroke and because of its restricted exhaust port area. The latter feature is hard to avoid with poppet exhaust valves.

Loop-scavenged engines generally have a little more than half a cylinder circumference available for inlet porting, and for the same bore/stroke ratio and port height can have a somewhat greater ratio of inlet port area to piston area than the opposed-piston engine. This particular advantage over the opposed-piston engine has not always been appreciated. Differences in details of port design account for the fact that this advantage does not appear in Fig. 12.

In general, while loop-scavenged types will have lower imep than through-scavenged types under comparable conditions, the output of loop-scavenged engines in proportion to weight and bulk may be no smaller because of the reduced ratio of engine size to cylinder capacity that is achieved by the elimination of the poppet-valve gear or the extra crankshaft, as the case may be.

Port Height – For a cylinder of given dimensions with well-designed ports, the only way to achieve a considerable increase in flow coefficient is to increase the port height, which means that the ports will open earlier in the expansion stroke and will remain open longer. If we start with the port design that gives the highest imep at a given scavenging ratio, it is obvious that such increases in height will reduce the imep at that scavenging ratio. Again, however, if the reduction in imep is less than the reduction in mep required by the pump, a net gain is achieved. Quantitative data on this point are not yet available, but the soundness of this conclusion is indicated by the very high ports found necessary in the GM-51 engine,[14] which is rated at an unusually high piston speed for a 2-stroke engine.

Cylinder Size – As in the case of 4-stroke engines,[15] 2-stroke engines obey the laws of similitude, and cylinders of the same design but of different size will have equal mechanical stresses and the same scavenging ratio and scavenging efficiency, when run at the same mean piston speed. This relation means that the advantage in size and weight for a given output will lie with small cylinders, as long as reasonable design similitude can be maintained.

Supercharging 2-Stroke Engines – Fig. 10 illustrates the futility of attempting to achieve large gains in output merely by raising the scavenging ratio of 2-stroke engines. This figure indicates that, as scavenging ratio is increased, a point will soon be reached where the resultant increase in scavenging efficiency is too small to justify the added mep required by the scavenging pump. This conclusion holds even for through-scavenged designs, which have a so-called "supercharging period," that is, a lag between exhaust closing and inlet closing.

The only way any considerable gain in imep can be achieved in 2-stroke engines by supercharging is to raise *exhaust pressure,* as well as inlet pressure, above that of the surrounding atmosphere. The use of an exhaust turbine immediately suggests itself.

Free-piston engines are the only current examples of 2-stroke engines with considerable super-

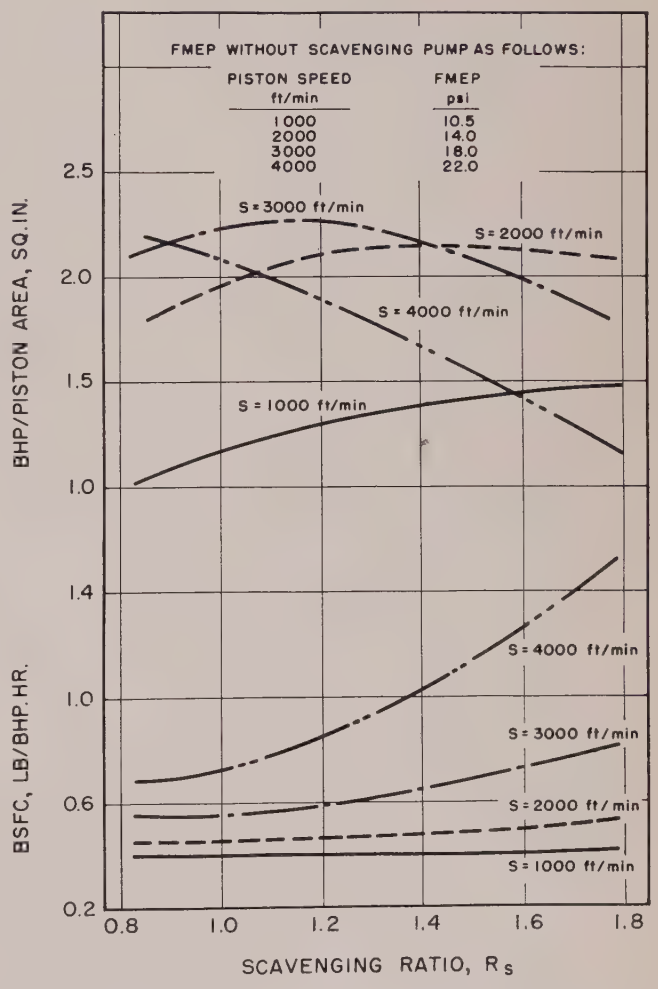


Fig. 13 – Computed performance of 2-stroke diesel engine at various scavenging ratios. Assumptions:

isfc = 0.344, $\eta_c = 0.75$, $C = 0.03$, $r = 16$, $T_e = 60$ F, $p_e = 14.7$ psia

[14] See *SAE Journal,* Vol. 61, June, 1953, pp. 48-55: "Loop Scavenging Used in New 3000-Rpm Diesel," by J. Dickson and R. D. Wellington.

[15] See ASME Transactions, Vol. 72, July, 1950, pp. 633-645: "Effect of Size on Design and Performance of Internal-Combustion Engines," by C. F. Taylor.

charging. If such engines prove to have good durability and reliability, this development should encourage the builders of conventional 2-stroke engines toward turbosupercharging.

Conclusion – It is hoped that the methods of measuring scavenging efficiency suggested here will encourage manufacturers of 2-stroke engines to measure this important quantity. It is also hoped that some of the basic relations here pointed out will be of help to future designers of 2-stroke engines.

APPENDIX I

$$R_s = \frac{M}{N\ \rho_s\ V_c} \qquad (2)$$

N = Rpm × No. of cylinders

Piston speed, s = 2 × rpm × stroke

so that:

$$\text{Rpm} = \frac{s}{2 \times \text{stroke}}$$

$$\text{No. of cylinders} \times V_c = \left(\frac{r}{r-1}\right) \times A_p \times \text{stroke}$$

where A_p = area of all pistons

$$R_s = \frac{M}{\left(\frac{r}{r-1}\right)\left(A_p/2\right) s\ \rho}$$

But:

$$M = A_p\ C\ c_s\ \rho_i\ \phi \qquad (5)$$

Combining:

$$R_s = 2\ C \left(\frac{r-1}{r}\right)\left(\frac{\rho_i}{\rho_s}\right)\left(\frac{c_s}{s}\right)\phi$$

$$\frac{\rho_i}{\rho_s} = \frac{p_i}{p_e}$$

since ρ_i and ρ_s are both based on T_i and $p_s = p_e$

$$R_s = 2\ C \left(\frac{r-1}{r}\right)\left(\frac{p_i}{p_e}\right)\left(\frac{c_s}{s}\right)\phi \qquad (6)$$

APPENDIX II

$$\text{Mep} = \frac{\text{Work/no. of cycles}}{V_d}, \frac{\text{in.-lb}}{\text{in.}^3}$$

Compressor work per min = $M\ c_p\ (T_i - T_a) \times 778 \times 12$

Compressor work per cycle = $(M/N)\ c_p\ (T_i - T_a) \times 778 \times 12$

$$(T_i - T_a) = \frac{T_a\ y}{\eta_c}$$

$$Cmep = \frac{M/N\ c_p\ T_a\ y \times 778 \times 12}{V_c \left(\frac{r-1}{r}\right)\eta_c}$$

$$R_s = \frac{(M/N)\ 1728}{\rho_s\ V_c} \qquad (V_c \text{ in in.}^3)$$

$$Cmep = \frac{R_s\ \rho_s \left(\frac{r}{r-1}\right) c_p\ T_a\ y \times 778}{\eta_c \times 144} \qquad (7)$$

DISCUSSION

Favor Single-Sample Method Used by Authors

– D. A. Harnsberger and F. S. Schaub
Cooper-Bessemer Corp.

WE have tried several different methods for determining scavenging efficiency of 2-stroke gas engines, and we agree that the single-sample method as presented in this paper is by far the most practical to apply and gives the most reliable results.

We have found it convenient to apply both in laboratory and field testing of 14x14 and 18x20 size engines.

One convenient use of the impact sampler in the field is simply to determine if the fuel/air ratio of a given design under field conditions is in a safe operational range.

In our laboratory testing one effect that has been noticed is that scavenging efficiency decreases slightly with increasing load. We should like to know whether the authors have noticed a similar trend, and whether this effect might be detrimental to getting a true picture of scavenging efficiency.

When developing a piston-ported 2-stroke design, should complete scavenging of the volume above the ports be the ultimate aim in efficiency? For example, if the ports of an engine occupy about 30% of the total stroke, does this not mean that a 70% scavenging efficiency would represent perfect scavenging of this design?

Referring to Fig. 8, is the spread in fuel/air ratio results between spark-ignited and diesel data due to more stratification under diesel operation or to differences in fuel composition? Have you experienced difficulty in obtaining uniform samples from open-chamber diesels?

To what degree is the increase in scavenging efficiency of the uniflow cylinder over the loop-scavenged cylinder a result of a more easily cleaned cylinder, and to what degree is it a function of supercharge made possible by unsymmetrical valve or port timing?

2-Stroke Engine Sensitive To Dynamic Effects

– Peter Kyropoulos
California Institute of Technology

IN 1952 the authors published a most important paper dealing with the breathing capacity of 4-stroke engines.[a] This paper was a summary of years of work.

The present paper does the same thing for 2-stroke engines.

A comparison of the number of references quoted in the two papers shows that the 2-stroke engine has received relatively little attention in this country. Extensive work has been done and published in Germany, which has been summarized by Schweitzer.[b]

The present authors have established a number of simple and physically sound parameters that characterize the breathing of 2-stroke engines. They then have conducted experimental work to establish variation of these breathing parameters with engine performance parameters as well as engine geometry.

For those who have tried to use published papers in

[a] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 617-636: "Volumetric Efficiency of 4-Stroke Engines," by J. C. Livengood, A. R. Rogowski, and C. F. Taylor.

[b] See "Scavenging of 2-Stroke-Cycle Engines," by P. H. Schweitzer. Pub. by Macmillan, New York, 1949.

designing of engines, the present paper will be a welcome contribution because:

1. Similarity considerations permit application of the data presented to other engines.

2. A careful evaluation is made of experimental methods used in obtaining the data.

We were particularly interested in the sampling valve technique of measuring scavenging efficiency.

In Fig. 10 we should like to know whether the large circle at $R_s = 1.4$ and $e_s = 0.7$ on the opposed-piston curve indicates the vertical spread of points.

Figs. 11 and 12, which show flow coefficient versus p_e/p_i for motoring, firing, and steady flow and for different engines, are most interesting. Such data are scarce, hard to get, and very valuable for class work.

The sharp increase in C at high values of p_e/p_i is not what one would expect to happen. We have suspected it on the basis of some experiments that we made on a large crankcase-scavenged engine. We believe that under these conditions dynamic effects become very strong and cause the abnormal behavior.

The paper is restricted to the engine itself. It would be well to emphasize the sensitivity of 2-stroke engines to the dynamics of intake and exhaust system.

Favors Simplified Definition of Terms

— A. W. Hussmann
Pennsylvania State College

THERE is a regrettable multitude of definitions and quite a confusion in the terminology when it comes to 2-stroke-cycle engines. M. I. T. has settled on its own terminology as given in the paper just presented, but other people also have their own, and so the confusion grows.

I agree that there is much logic in the definition of the scavenging ratio, scavenging efficiency, and charging efficiency as presented here by using a well-defined ideal charge as criterion and as base value. However, much can be said in favor of more simplified definitions as, for instance, Schweitzer has already given.[b] To be more specific: These authors define the scavenging ratio as the air delivered per stroke to the amount of air that would fill the total cylinder volume under inlet manifold temperature and exhaust manifold pressure. To define that ratio one has to know not only the air delivered and the displacement of the engine but also the compression ratio, the inlet manifold temperature, and the exhaust manifold pressure—data that are frequently not available and that vary themselves with speed and load. In Schweitzer's definition the scavenging ratio or, perhaps better, the delivery ratio (German "Liefergrad") is simply the volume actually delivered at normal temperature and pressure conditions over the displacement of the engine. In analyzing the scavenging characteristic of a particular engine under various conditions, the definition of the present authors is preferable. However, the practical engineer is looking for a simple, more readily acceptable definition and he seems to prefer, as far as I have seen in diesel-engine practice, the simpler terms relating volumina to the known displacement volume. It may not be scientific, but it is, nevertheless, practicable.

I wish to compliment M.I.T. on the development of the impact sampler. With such a simple automatic sampler, it now becomes possible for the first time to make field tests and to obtain the much needed information on trapping efficiency and actual air/fuel ratios during the combustion process.

The concept of treating a 2-stroke-cycle cylinder as a constant orifice is generally accepted today. The flow coefficient as defined in equation (5), however, loses its meaning as flow constant and is degraded to a mere scale factor if the piston area is introduced as the orifice area. True, for a given engine the equivalent orifice area is proportional to the piston area. This proportionality factor, however, is significantly different for various types of scavenging systems and even varies for the same type of system with the geometry of ports or valves and with the timing. It would have been most useful to have numerical information of the actual flow coefficient during the blowdown and scavenging process for a detailed analysis. This, of course, would be a sizable research project in its own right. However, we could go at least half-way by defining as orifice area the reduced area, formed by considering inlet and exhaust time areas as restrictions in series. For all diesel engines for which we could get hold of enough data, the "flow coefficient" came out to be in the close neighborhood of 0.85. It is, of course, still not a true flow coefficient and we might do better to invent another name for it (Schweitzer calls it "scavenge factor"). For the benefit of those who want to apply the very valuable information contained in the flow coefficients presented in Figs. 11 and 12, it would be helpful or even necessary to know not only the piston area but also the valve timing diagrams for the engines investigated. Publication of these data along with the flow coefficients would be much appreciated by many of us.

This Type of Analysis Should Be Used More Often

— Blake Reynolds
Texaco Development Corp.

THIS paper presents a down-to-earth analysis of the scavenging process, a thoroughly practical description of ways to measure scavenging efficiency, and reliable numerical data on the scavenging performance of several well-developed engines. These elements are combined to give a broad understanding of the scavenging process and to show how scavenging performance characteristics can be used to advantage in design and development.

The method of analysis itself is logical, simple, and workable. It is based on two idealized cases, the scavenging process and the flow process, which are well-reasoned and free of extraneous complications. For instance, the ideal volume for scavenging considerations is taken as the total cylinder volume. Clearance volume is included because the scavenging process is unaffected by the previous top-center position of the piston, there being no relation between the clearance volume and the residual gas, as in the ideal 4-stroke case. Also, the piston displacement below port closing is retained, just as in the 4-stroke case, for several sound reasons:

1. Port closing has no absolute significance. It is not generally the time of zero pressure drop across the port.

2. The goal is to utilize the full volume to the best advantage, not just the volume above port closure.

3. A variable base would make comparisons between engines misleading, for a higher scavenging efficiency if based on a smaller volume is not necessarily an improvement.

Another example of a sound practical approach is the use of exhaust instead of intake pressure in the term scavenging density.

These ideal cases lead to three factors, scavenging ratio, scavenging efficiency, and port flow coefficient, which adequately represent the inherent scavenging characteristics of any given 2-stroke engine. The factors allow for the variation of all operating variables while maintaining a common base. Thus, they may be used for comparing the scavenging performance of different engines as well as for predicting the scavenging performance in untested

operating or design areas. Furthermore, since the factors give retained air and blower power, they are directly useful for predicting brake power and fuel consumption.

This type of analysis, even this specific method, is not new, but unfortunately it appears not to be generally practiced in 2-stroke development. Perhaps the greatest single deterrent has been a fear of attempting to measure the scavenging efficiency, without which any analysis is crippled. In any event, the thoroughly practical discussion on the measurement of scavenging efficiency is a valuable portion of the paper and should encourage more general use.

By way of demonstrating the application of these methods, the paper includes numerical data on the scavenging performance of engines representative of the three common 2-stroke types. This information is a gold mine in itself for the 2-stroke engineer. It gives him both a practical yardstick by which to measure his own development and a reliable foundation on which to base design estimates.

Authors' Work Has Great Practical Significance

– P. M. Ku
National Bureau of Standards

IN this paper, the flow of scavenging air through the 2-stroke cylinder is treated as flow through an orifice. The method of treating the cylinder as one orifice provides a simple and convenient means for the overall study of scavenging ratio. On this basis, the authors have thrown much light upon the mechanism of the flow process in a general sense. However, for detailed investigation of the flow process, a knowledge of the inlet and exhaust ports as individual orifices in series appears to be essential. Indeed, it would be interesting if an analytical solution based upon thermodynamic and fluid-dynamic laws can be worked out for this process, as the M.I.T. group has so skillfully done for flow through the 4-stroke cylinder.

There is at present a lack of basic understanding of the mechanism of the mixing and retaining process in the 2-stroke cylinder. This is due not only to the complexity of the problem itself but also the difficulty in accurately measuring the quantity of the retained air. In this connection, the reported work on scavenging efficiency determination has great practical significance. Fig. 6 shows that the relationship between scavenging efficiency and scavenging ratio is influenced by engine speed. It is entirely possible that such other factors as inlet and exhaust pressures, inlet and exhaust port areas, and port timing also enter the problem. With such tools as reported in this paper, perhaps the effect of design and operating variables on scavenging efficiency can be more fully explored in the near future.

In the meantime, it appears that research aimed at providing a basic understanding of the mixing and retaining process is indispensable. The literature records several attempts at qualitative observation of the flow pattern in the 2-stroke cylinder, without resorting to quantitatively defined terms. It would be a distinct step forward to introduce quantitative concepts to such research. An investigation of this type is currently under consideration at the National Bureau of Standards.

Arbitrary Concepts Make Data Hard to Adapt to Design Purposes

– W. A. Pullman
Pennsylvania State College

UNFORTUNATELY it would appear that the data presented by these authors cannot be readily adapted for design purposes since the authors have employed arbitrary concepts, such as the flow coefficient C.

This practice is open to objection since the flow coefficient, as defined, is not only a function of the port geometry, but includes the extraneous effects of incomplete blowdown and the influence of wave motion in the intake and exhaust pipes when these pipes are of a significant length.

Would it not, therefore, have been better to express the flow under steady-flow conditions in terms of a conventional coefficient of discharge for each port at various port openings.

Further, the coefficients so obtained should not be averaged over the whole cycle, since, in the case where wave motion is present in the intake and exhaust pipes, serious errors would arise. For efficient scavenging it is desirable to have a rarefaction wave arriving at the exhaust port at about bdc and for this wave to be most beneficial, it is essential that the greatest effective exhaust port area be available. The increase in airflow resulting from such a rarefaction wave would be masked in a calculation that employed mean values of port area and discharge coefficient.

With this preliminary knowledge of the steady-flow discharge coefficients the blowdown and scavenging process of a running engine may be analyzed along the lines described by Wallace and Mitchell[c] and also by Cole and Mills[d], by considering the process as a succession of steadyflow intervals.

In this way it is possible to ascertain the part played by the various factors. For the dimensions of a 2-stroke cylinder that the writer has available and a given release pressure, such an analysis shows that, if the blowdown is complete at intake open at 1000 rpm, then at 2000 rpm the cylinder pressure at intake open is still 40% of its initial value, whilst for a speed of 300 rpm the blowdown occupies only 55% of the lead.

This trend bears out the authors' statement regarding the decrease of the flow coefficient C with increase in speed under firing conditions. Would the authors offer an explanation for the slight but consistent increase in the value of C with speed for the motoring tests?

2-Stroke Engine Is Tough to Analyze

– K. L. Hulsing
Detroit Diesel-Engine Division, GMC

THE authors have defined their terms, ideal scavenging, scavenging ratio, scavenging efficiency, and charging efficiency. It is well that they have done this because there is much disagreement in the literature as to what these terms actually mean. While we may not be in complete agreement with their definition, at least we know what they are trying to do.

One of the methods proposed for determining scavenging efficiency by the use of an impact sampler appears to be a very worthwhile development. It is simple and should give good results. It appears to be the type of instrument that industry can use to supplement its normal laboratory instrumentation.

The 2-stroke engine, particularly the loop-scavenged type, is one of the simplest engines to look at and yet one of the toughest to analyze and develop. The reason for this is probably the fact that independent factors cannot be

[c] See Institution of Mechanical Engineers, Proceedings (B), Vol. 1B, No. 8 – General, 1952-53, pp. 343-363: "Wave Action Following Sudden Release of Air through an Engine Port System," by F. J. Wallace and R. W. Stuart Mitchell.

[d] See Institution of Mechanical Engineers, Proceedings (B), Vol. 1B, No. 8 – General, 1952-53, pp. 364-378: "Theory of Sudden Enlargements Applied to Poppet Exhaust Valve, with Special Reference to Exhaust-Pulse Scavenging," by B. N. Cole and B. Mills.

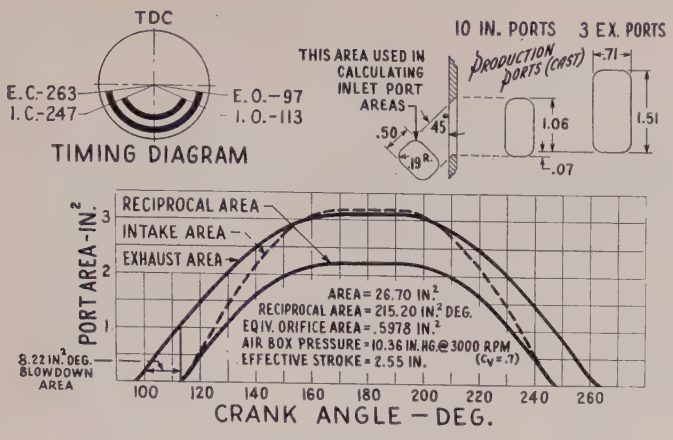


Fig. A – Cylinder liner time-area diagram for 4-51 engine

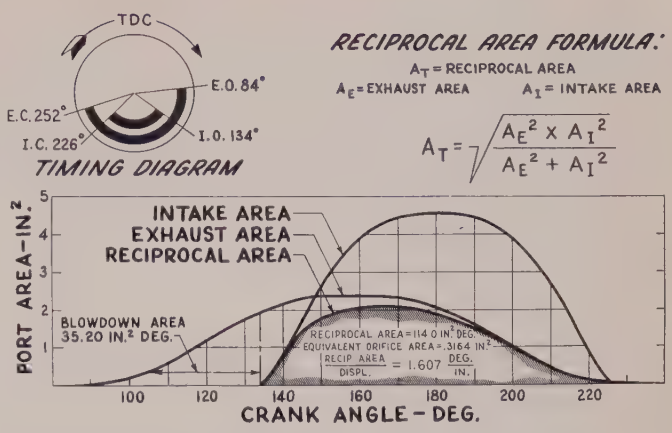


Fig. B – Time-area diagram for series 71 production cylinder – 1950

controlled. To illustrate my point, let us assume that the development engineer decides to increase the height of the exhaust ports in the loop engine. Immediately he has changed the engine timing and the ability of the engine to trap air. In addition, the compression ratio has been lowered and the blowdown time area has been increased.

As you can see, by this simple change he has affected five other factors that influence the ability of the engine to produce power. If we now assume that we are going to develop this engine with a selected compression ratio, it will be necessary to change the combustion chamber. This may be done by decreasing the piston-to-head clearance, decreasing the diameter of the bowl or the depth of the bowl, or any combination of these, all of which probably will affect the combustion process. Before we know it we probably will be changing the injector tip or the spray angle or the number of holes. It is readily apparent that before long a person reaches the conclusion that he is chasing himself around a circle.

The importance of being able to determine the equivalent orifice size of an engine soon becomes apparent when working on the 2-stroke engine. It permits an engine evaluation to be made and power requirements of the scavenging blower to be computed before a given design is released for test. We use what we call the reciprocal area method of determining the orifice area of an engine. This method consists primarily of reducing the blow-through time area of the engine to an area indicated to be a reciprocal area. (See Fig. A.) This area is expressed as in.²-deg, and if we now divide this by 360 deg the result will be in.² Then, by plotting an airflow function as a constant and selecting a suitable flow coefficient, it is possible to calculate with reasonable accuracy the pressure differential necessary to flow any given amount of air through the engine. Once the pressure differential has been established and the scavenging ratio has been selected (usually from experience) the horsepower demands of the scavenging pump can readily be determined.

The authors have indicated in their paper that the blowdown area of an engine greatly affects its flow coefficient. We have been unable to attach too much significance to the blowdown area as it affects the engine orifice flow coefficient. We have already looked at Fig. A, which is the time area diagram of our 51 series engine. You will notice that the blowdown time area is 8.22 in.²-deg and the reciprocal time area is 215.20 in.²-deg, which gives a ratio of 3.8%. On our series 71 engine (Fig. B.), we have 35.20 in.²-deg blowdown and the reciprocal area of 114 in.²-deg, or a blowdown ratio of 30.8%. The 51 engine at 3000 rpm has a piston speed of 2050 fpm. The 71 series running at 2100 rpm has a piston speed of 1750 fpm.

It would seem that, from a rational point of view, these two engines should have their breathing systems from a blowdown standpoint interchanged.

The reference of the authors to the desirability of having a large stroke/bore ratio to obtain maximum port area is a theoretically correct statement, and I am sure that their intention was to convey this impression. Actually, of course, from many design considerations and also scavenging considerations, a stroke/bore ratio of approximately one would appear to be an advantage. In the design of the loop-scavenged engine, because of the fact the bmep's calculated on the basis of displacement are necessarily low, it becomes mandatory to design the 2-stroke loop-scavenged engine with the maximum displacement possible within a given framework, so as to make the final physical size comparable to engines using other scavenging systems.

Authors' Closure To Discussion

PROF. ROGOWSKI: In reply to Mr. Harnsburger's questions, we have not measured scavenging efficiencies of enough engines in which load is controlled by fuel/air ratio to say definitely that scavenging efficiency always decreases slightly with load. It seems reasonable, however, that this would be the case since higher fuel quantities will result in higher cylinder pressures at the time of inlet opening, and therefore more blowback into the inlet system. Since scavenging efficiency is a measure of air retained, or potential maximum power, it is probably most important to measure it under rated power conditions.

We feel that scavenging the entire cylinder volume is the most useful ideal. The volumetric efficiency of 4-stroke engines, for example, is not based on the volume above the piston when the inlet valve closes. If the amount of air trapped (scavenging efficiency) suffers because the ports are too high, this may be justified if the scavenging power requirements are sufficiently reduced. Otherwise the design should be changed.

Referring to Fig. 8, the difference in carbon-hydrogen ratio between diesel and spark-ignition fuels is too small to affect the curves. We believe the difference is mostly due to the lower "combustion efficiency" of the compression-ignition engine, because of the comparatively poorer mixing and evaporation of the fuel. Measuring the cylinder fuel/air ratio by the method of Gerrish and Meem[10], in which all of the combined carbon in the exhaust is oxidized in the analyzing apparatus, should avoid any error from this source except when considerable free carbon is present in the exhaust.

From somewhat limited data, we feel that most of the scavenging advantage of the uniflow engine is due to the favorable flow pattern.

In answer to Prof. Kyropoulos' question about Fig. 10, the large circle merely indicates that the three cylinders all have exactly the same scavenging efficiency for this scavenging ratio.

With regard to Dr. Pullman's query regarding the effect of piston speed on motoring port coefficient, it appears that in Fig. 11a there is an increase in C with speed, while in Fig. 11b, the smaller engine shows a reduction in C with speed. These effects are small and may be due to pressure waves. The piping was short but not geometrically similar in the two engines.

PROF. TAYLOR: I would like to add only a brief discussion of some of the reasoning behind our definitions and methods of approach.

Some years ago, I suggested the formation of an SAE committee to formulate 2-stroke nomenclature, but since this was not done, we have had to choose terms and definitions that seemed to us to be both convenient and significant. These definitions were chosen with a view toward quantitative evaluation of the performance of existing 2-stroke engines, and prediction of the performance and scavenging requirements of new designs. The subject of individual port design, timing, and so on, has not been considered in this paper, except in the most general terms.

The use of volumetric or scavenging efficiencies based on standard air density would not measure cylinder performance for engines tested under other-than-standard conditions. On the other hand, a scavenging density based on inlet temperature and exhaust pressure measures cylinder performance as a function of the engine's actual environment at the time of the test, and thus seems, at least to us, far more significant.

The assumption that the effective cylinder volume is only that above port closing ignores dynamic effects, and could lead to the grotesque conclusion that a cylinder with the ports extending for its whole length would have 100% scavenging efficiency! Since the size of the cylinder depends on the whole stroke plus the clearance space, it seems best to base all definitions on piston displacement and clearance volume, as in the case of 4-stroke engines – even those that have a very late inlet-valve closing.

The flow of gases through the cylinder, in a 2-stroke engine, is a very complex thing and we have not attempted to discuss it in detail in this paper. Flow coefficients for individual ports are important in port design, but the quality of the whole porting system depends not only on the individual flow coefficients of the ports, but on their size and arrangement with respect to each other and with respect to the piston and cylinder-head configuration.

The best way to measure this overall result is by actual engine test, and a convenient way to present the results of such measurements is in the form of flow coefficients based on some important area connected with cylinder size, such as the piston area. To express what I mean in another way, it is obvious that a perfect flow coefficient for each individual port will not result in good engine performance if the total port area is too small compared to the piston area. A flow coefficient based on piston area takes into account shape, size, and timing of the whole system, in so far as these items affect the flow through the engine during the scavenging period. This coefficient can be converted to any area base desired, simply by multiplying it by the ratio of piston area to the new base area.

In considering the problem of 2-stroke engine design, it is well to remember that 2-stroke practice is still in a very fluid stage, as evidenced by the wide range of port designs now in use. If this paper stimulates renewed efforts to improve our knowledge in this important field, it will have served its principal purpose.

ORAL DISCUSSION

Reported by Gregory Flynn, Jr.

Research Laboratories Division, GMC

Dr. Taylor: Our method of scavenging analysis enables comparison to be made between engines of widely different design. It is important to compare engines on the basis of piston speed rather than rpm.

E. L. Dahlund, Fairbanks, Morse, & Co.: Scavenging efficiency is not all that the engine designer desires. Scavenging air provides a good deal of necessary cooling as well as more bmep. There are as many port designs as there are designers, and we should remember that port design (port bridges) has a marked effect on cylinder life.

W. H. Percival, Research Laboratories Division, GMC.: I have done some work along the lines indicated in the paper, and I consider it the best approach to the scavenging analysis problem. Our attempts to use the exhaust-gas sampling valve discussed in the paper have, however, proved unsuccessful.

Dr. Taylor: I would like to stress the need for agreement on definitions used in 2-stroke-engine work. Some time ago I requested the SAE to set up a committee to agree on definitions. The definitions used in this paper were adopted for convenience, and if any definition is based on standard conditions that change with the weather, then change the definition. Thus, it is much more desirable to make all parameters dimensionless.

Our work was intended to show general relationships and not the effect of detailed design. Thus, a short-stroke engine that penalizes the scavenging may be a justifiable sacrifice from an overall engine design standpoint. What the M.I.T. method drives at is an evaluation of the resistance to flow of the engine with respect to piston area, neglecting port design and port coefficients. The parameters used are then valuable for general comparison of the engine design as a whole.

R. Miller, Nordberg Mfg. Co.: What is the explanation for the difference in scavenge efficiency between the through-scavenged engine and the opposed-piston engine?

High scavenging efficiency with low blower input horsepower points to turbosupercharging as the answer to 2-stroke design problems.

Engine efficiency depends on expansion ratio and not compression ratio, as is popularly believed.

Mr. Rogowski: The difference between the uniflow and the opposed-piston engine was probably just due to engine design and the fact that the opposed-piston engine was a very old model. I agree that turbosupercharging is the answer to higher output.

M. A. Elliott, Illinois Institute of Technology: With fuel/air ratios of 0.05 to 0.06, as shown in the paper, errors can be introduced by the small quantities of unburned carbon in the exhaust.

Mr. Rogowski: I do not know if any unburned carbon was present.

F. G. Shoemaker, Detroit Diesel-Engine Division, GMC: One sure, quick way to determine if more oxygen is trapped in the cylinder is to open the throttle wide and see if more power is produced. Another good way to check scavenging is by skip cycling, that is, injecting fuel only once every four or six cycles on a 2-stroke engine.

R. J. McCrory, Battelle Memorial Institute: Since very little is known of the exact mechanism of scavenging, I would suggest dynamic model tests. Have the authors considered such methods?

Mr. Rogowski: Steady-flow model work is useless, for no correlation can be made with the engine.

Compression and End-Gas Temperatures from Iodine Absorption Spectra

S. K. Chen
International Harvester Co.

N. J. Beck
Douglas Aircraft Co.

O. A. Uyehara and P. S. Myers
University of Wisconsin

This paper was presented at the SAE National Fuels & Lubricants Meeting, Chicago, Nov. 6, 1953.

THIS paper discusses a method of measuring compression temperature by means of the absorption of light.

An optical-electronic system measures the change in color of a trace of iodine gas that has been added to the intake mixture. From these measurements the temperature of the iodine and by inference, the temperature of the gases, is determined.

The apparatus used is described briefly and the test results obtained in measuring compression and end-gas temperatures in a spark-ignition engine are also presented.

THE University of Wisconsin has been engaged in engine combustion research for a number of years. Much of the work has involved a correlation between the measured pressures and temperatures in an engine. These measurements have offered some insight as to the actual phenomena occurring during the complex combustion processes.

Previously, there had been no satisfactory technique for unsteady-state determination of combustion-chamber gas temperatures below about 2000 F. Consequently, a program was initiated to develop a technique for measuring gas temperatures in the range of compression and end-gas temperatures in a spark-ignited engine, on the premise that such measurements could supply useful information for analyzing precombustion reactions, weights of charge and residual gas, changes in number of moles, and other combustion phenomena.

The various methods of temperature measurement that were considered were (a) thermocouples, (b) resistance thermometers, (c) determination of velocity of sound, (d) radiation of light, and (e) absorption of light.

The time rate of response of both thermocouples and resistance thermometers was not considered adequate and their susceptibility to surface reactions was considered undesirable. The velocity-of-sound technique was considered but, since it offered experimental difficulties and required a knowledge of the ratio of the specific heats, k, it was decided to investigate other means. Subsequent to this decision it was found that this technique was being investigated with considerable success by J. C. Livengood and C. F. Taylor at Massachusetts Institute of Technology. The intensity of radiation from the unburned gases was considered too weak to allow practical measurements, although later experimental work at the University of Wiscon-

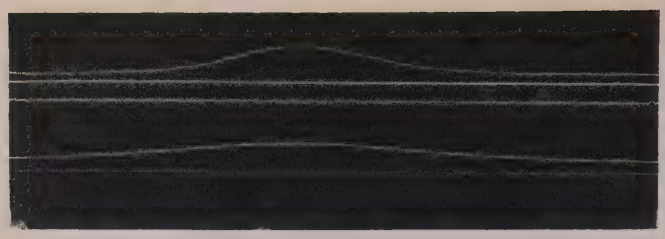


Fig. 1 – Voltage-time output of lead sulfid cell receiving radiation from motored engine

sin[1] indicates that such a technique might be possible. Fig. 1 presents a curve of radiation versus crank angle as obtained from a motored CFR engine. Although this preliminary study suggests experimental difficulties in the use of this technique, work is continuing on this technique using the principles suggested by Millar, Uyehara, and Myers.[2]

After due consideration it was decided to try a light-absorption technique for measuring compression temperatures. The technique involves the use of a trace of iodine gas to indicate temperatures. An optical-electronic system measures the change in color of the iodine, from which measurements the temperature of the iodine and, by inference, the temperature of the gases is determined.

This paper briefly describes the apparatus used and the test results obtained in measuring compression and end-gas temperatures in a spark-ignition engine.

Theory

All practical temperature measuring devices involve the determination of some physical property that varies with temperature. Correlation of a measurement of this physical property of matter with temperature will produce a workable thermometer. The technique discussed here involves the correlation between temperature and the observed absorption spectrum of a gas.

The theory that predicts the effect of temperature on the light-absorbing properties of gases is well established and has been verified experimentally. The principles involved are basically quite simple and can be described in a simple manner by the following example: When white light passes through iodine gas, the emerging beam changes color from a deep red to light blue when the iodine gas is heated from room temperature to about 1300 F. This change in color of the emerging light results from the change in the relative proportions of blue and red light that are absorbed by the iodine vapor at the two temperatures. By measuring this effect quantitatively it is possible to measure the temperature of the iodine gas. The theory and experimental techniques have already been described in detail.[3] The effects of temperature on the absorption spectra of gases are not peculiar to iodine gas but can be observed in any gas that has a measurable absorption spectrum. However, of all the gases considered, iodine seemed to offer the best compromise of properties for practical temperature determinations.

The spectral regions used for temperature determinations are indicated in Fig. 2. This is a photograph of an iodine absorption spectrum taken using an incandescent light source. The dark bands throughout the spectrum correspond to light absorption. (The dark region at the long wavelength end of the spectrum is caused by decreased film sensitivity and not by light absorption.) The relative proportions of light absorbed by the regions I_0, I_1, and I_2 vary with temperature. For example, an increase in temperature causes a de-

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[1] "Radiation Measurements with Lead Sulfid Cell Using Spark-Ignited Engine," by D. A. Trayser. M.S. thesis, University of Wisconsin, 1953.

[2] See "Practical Application of Engine Flame Temperature Measurements," by G. H. Millar, O. A. Uyehara, and P. S. Myers, on pp. 514-530 of this volume.

[3] See Appendix 1.

crease in the intensity of the I_0 absorption bands and an increase in the intensity of the I_2 bands. Negligible absorption occurs in the region denoted by I_{tot}. The intensity of light in the I_{tot} region is used only as a reference for determining whether the intensity of the light source as seen by the phototube has changed either due to a change in the temperature of the source or to a change in absorption of the elements in the light path.

The steady-state calibration (or correlation of the color of the emerging light with the temperature of the iodine) covered the range of temperatures and iodine concentrations that were expected to be covered in actual use in an engine. The calibration procedure made it possible to determine both temperature and iodine concentration from absorption measurements.

Use of the absorption technique for measuring temperatures in an engine required the provision of a light path through the combustion chamber and necessitated the addition of a trace of iodine to the intake mixture. The quantities of iodine required were of the order of 0.02% by volume.

It should be borne in mind that the temperatures determined from the iodine absorption spectrum correspond to the vibrational temperature of the iodine gas. However, barring the effects of anomalous reactions, the iodine gas should attain equilibrium with a mixture of gases in a time interval of the order of 10^{-8} seconds,[3] which, for most purposes, can be considered to be instantaneous. Observations of the absorption spectrum therefore should provide a means for instantaneous and continuous temperature measurements of a mixture of gases.

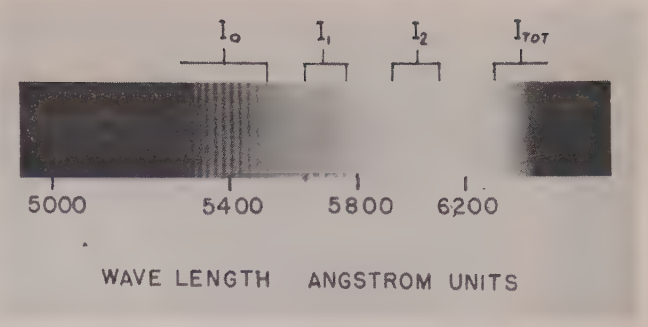

Fig. 2 – Iodine absorption spectrum – absorption bands identified as follows:

I_0 – Ground state

I_1 – First vibrational energy level

I_2 – Second vibrational energy level

I_{tot} – No absorption

Apparatus and Experimental Procedure

The apparatus used consisted of an iodine vaporizer, a CFR L-Head engine, an absorption pyrometer, and associated recording equipment. A schematic diagram of the absorption pyrometer is shown in Fig. 3. Its operation is as follows: Light passing through an absorption cell (or engine combustion chamber) is focused on the slit of a spectrograph. The desired wavelength regions of the resulting spectrum are focused on phototubes and the appropriate electronic equipment produces an oscilloscope deflection that is proportional to light absorption at three different wavelengths. The operation of the absorption pyrometer has been described in detail in an earlier reference.[3]

The engine used was a CFR L-Head engine with a compression ratio of 5.6/1. The cylinder head and block were modified to provide a light path through the combustion chamber. Fig. 4 is a photograph of the cylinder head showing the location of the quartz windows. The center opening in the cylinder head was used for the pressure indicator and the spark plug was located in the opening farthest from the light path. This particular location of the spark plug was used in an attempt to cause the end gas to be located in the absorption path. The quartz window was a 3-in.-long piece of $\frac{3}{8}$-in. fused quartz rod and was installed in a holder similar to the self-cleaning window described in an earlier paper.[4]

Iodine vapor was introduced into the engine intake system ahead of the carburetor. The iodine

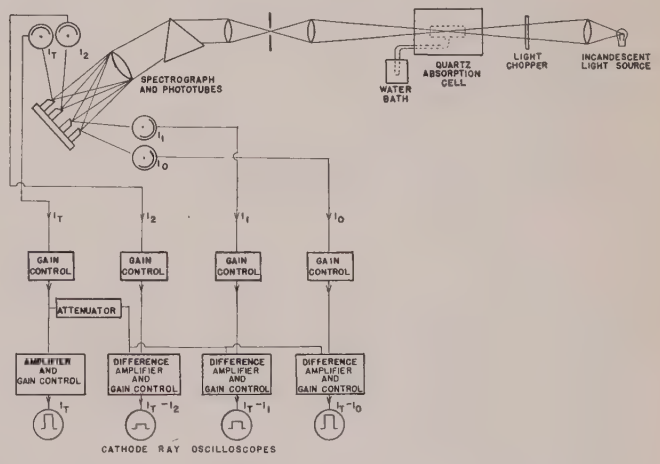

Fig. 3 – Schematic diagram of absorption pyrometer

[4] See ASME Transactions, Vol. 69 July, 1947, pp. 465-477: "Diesel Combustion Temperatures – Influence of Operating Variables," by O. A. Uyehara, P. S. Myers, K. M. Watson, and L. A. Wilson.

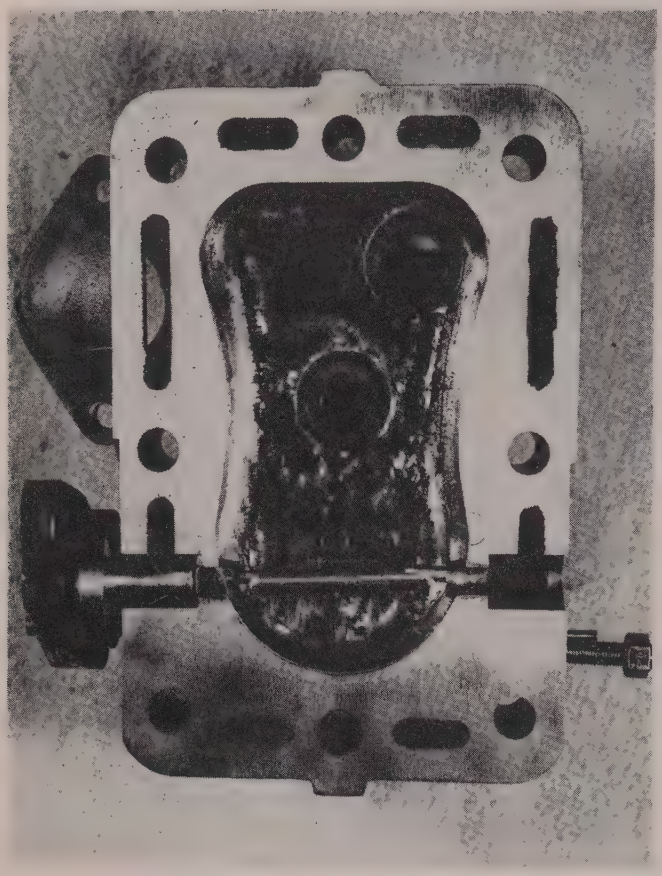

Fig. 4 – Photograph of modified cylinder head showing location of quartz windows

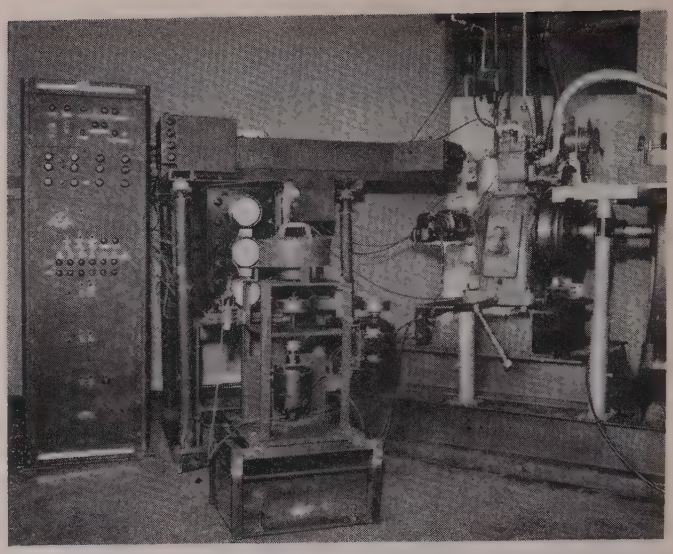

Fig. 5 – Photograph of engine and instrumentation

vaporizer was a stainless-steel tank enclosed in a furnace operated at about 350 F. Compressed air was bled into the vaporizer and the outlet mixture of air and iodine vapor was then fed into the engine intake. This method permitted adequate control of iodine concentration and allowed the iodine to be introduced only when needed for temperature measurements. The arrangement of the apparatus for engine temperature measurements is shown in Fig. 5.

Data were recorded by means of a drum camera driven at camshaft speed by means of a pair of selsyn motors. Three dual-beam oscilloscopes were mounted in a vertical plane. No horizontal deflection of the beams was used. An $f/1.9$ lens formed an image of the oscilloscope beams along a vertical line on a 30-in. strip of film (size No. 620) wrapped around the camera drum. Horizontal motion of the traces was thus provided by the motion of the film, resulting in an expansion of the time axis to about 25 crank angle deg per in. of film. The vertical deflections provided simultaneous records of combustion-chamber pressures, flywheel timing marks, and light absorption. The intensity controls on the oscilloscopes were triggered electronically such that the scope beams were turned on for only one engine cycle, thus providing complete pressure, temperature, and time records for a single engine cycle. The traces on the film strips were measured on an optical comparator capable of reading deflections to 0.0001 in. Special care was exercised to obtain narrow lines on the film so as to have a minimum of scaling error.

Sample oscilloscope traces are shown in Fig. 6. Reading from top to bottom the eight traces on the film represent:

1. Lamp intensity with no absorption. (Iodine gas does not absorb light of this wavelength.)

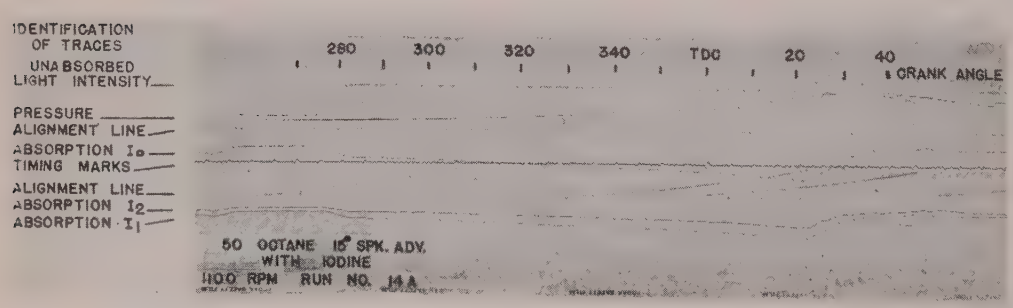

Fig. 6 – Sample photograph of oscilloscope traces with iodine in engine combustion chamber

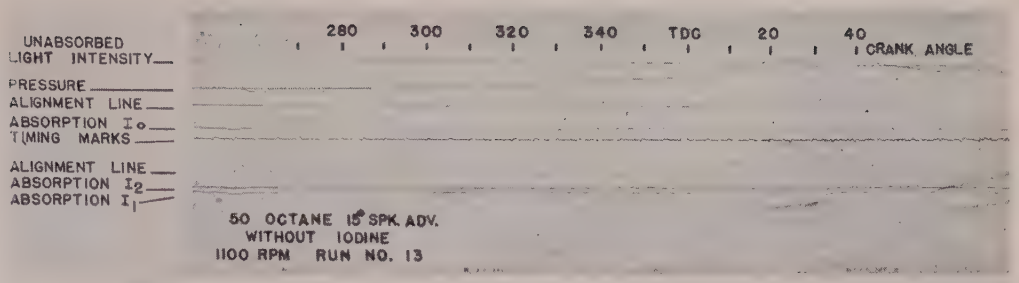

Fig. 7 – Sample photograph of oscilloscope traces without iodine in engine combustion chamber

2. Pressure, as measured by a Li-Draper strain-gage pressure indicator. (The step in the right-hand side of the pressure trace is a calibration signal equal to 41 psi.)

3. An alignment light located between the 'scopes, which provides a reference line for measuring deflections.

4. Light absorption at an average wavelength of 5230 A. (Absorption is indicated by a downward deflection.)

5. Crank-angle position giving one pip for every crank-angle degree. The pip was obtained by means of a magnetic pickup mounted near the engine flywheel and was caused by the standard flywheel markings of the CFR engine.

6. Another alignment light.

7. Light absorption at an average wavelength of 6050 A.

8. Light absorption at an average wavelength of 5750 A.

Experimental Data

Engine data were obtained primarily to determine if the technique of using iodine gas to indicate compression temperatures would prove practical. The engine was run at various operating conditions to aid in evaluating the iodine absorption technique and to obtain preliminary data on the effect of operating variables on the measured compression temperatures. The data presented here were obtained early in 1952. Additional results obtained about one year later[5] have confirmed the original data.

Data were obtained at arbitrarily selected operating conditions, which included two different speeds and spark settings, and three fuels with different octane ratings. These conditions were: 5.6/1 compression ratio, 212 F jacket temperature, 13.5 air/fuel ratio, engine speeds of 1100 and 1600 rpm, full throttle, 15- and 30-deg spark advance, and 50-, 70-, and 90-octane primary reference fuels.

The first engine data that were taken were for a run with no iodine introduced into the intake manifold to see if absorption was occurring from substances other than iodine. Fig. 7 shows an oscillogram obtained under such conditions. The fact that no deflection of the oscilloscope traces occurred previous to the arrival of the flame front under these conditions furnished experimental proof that absorption by substances other than iodine did not take place. As an additional experimental check, oscillograms similar to the one shown in Fig. 7 were taken at the end of each run. Negligible absorption was found under all conditions and fuels tested.

Figs. 8, 9, and 10 illustrate the consistency (and inconsistency) of temperature data obtained from the instrument and engine. The subscripts on the temperatures denote which combination of two wavelengths was used to compute the temperature. Fig. 8 presents typical data obtained under no-knock or light-knock conditions with all electronic instrumentation in good order and with the photographic conditions adjusted to give a maximum scaling accuracy. Fig. 9 shows the type of data obtained when a scaling error was made and an incorrect reading was obtained for the oscillograph deflection with zero light absorption. Such

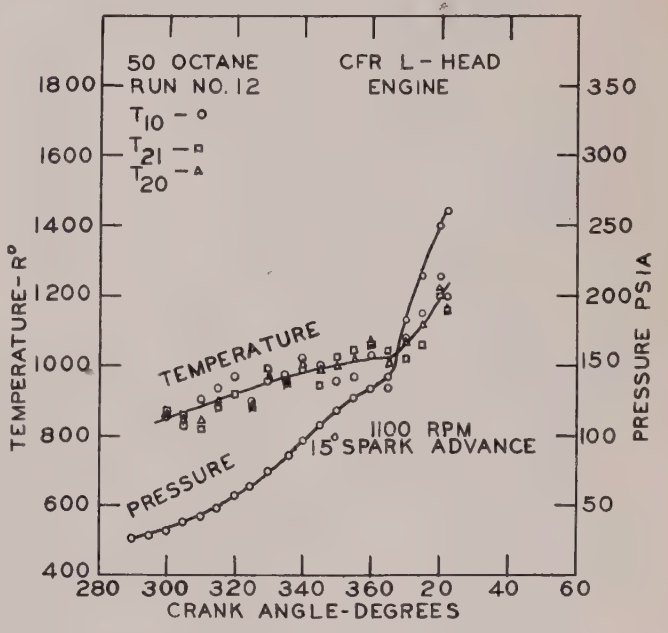

Fig. 8 – Typical engine data showing temperature and pressure as function of crank angle

[5] "Octane Numbers and Engine Preflame Temperatures and Pressures," by E. Freudman. M.S. thesis, University of Wisconsin, 1953.

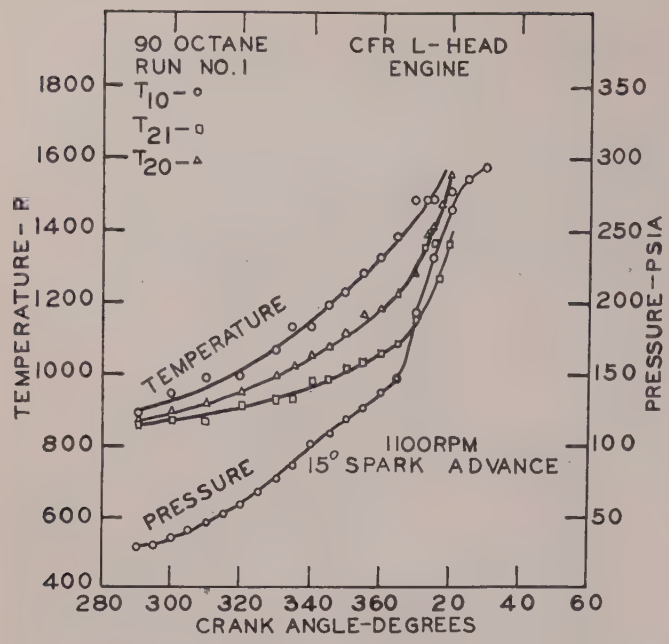

Fig. 9 – Data obtained when zero absorption line was incorrectly determined

data could usually be made more consistent by a recheck of the deflection corresponding to zero light absorption. Fig. 10 shows data where the agreement between the three temperatures is so bad that an average temperature cannot be determined. While such data would be obtained occasionally under any operating conditions, they would be most often obtained under engine conditions causing heavy knocking. It was noted that when such readings were obtained it was usually impossible to obtain a consistent zero absorption reference reading on the exhaust stroke of the engine. The reason for this has not been established, although it could be caused by the transmissivity of the quartz windows varying throughout the cycle. Data such as are shown in Fig. 10 were discarded.

The curves in Figs. 11 and 12 are presented to show the reproducibility of results. Each plot in these figures contains data from two separate engine cycles. The temperature values shown are the averages of the three temperatures determined from the three different combinations of wavelengths. The plotted points are the results of measurements made from the film strips at 5-deg. increments.

The quantity W/W_0 plotted on the curves is the ratio of the weight of charge in the cylinder divided by the weight of charge (at inlet conditions) per displacement volume. The weight W was computed from P, V, and T data as measured at the crank angle in question. W_0 was obtained from the inlet air pressure and temperature, the displacement volume, and an arbitrary factor of 0.95 to account approximately for the weight of clearance gases. The resulting W/W_0 can be considered analogous to volumetric efficiency, and should therefore be constant during any one cycle if the gas is homogeneous and of constant composition.

Figs. 13 and 14 present typical data for individual cycles taken at 1600 rpm. These typical cycles are presented for comparison with the data taken at 1100 rpm and shown in Figs. 10, 11, and 12.

By trial plots it was found that data obtained at 15- and 30-deg spark advances could be com-

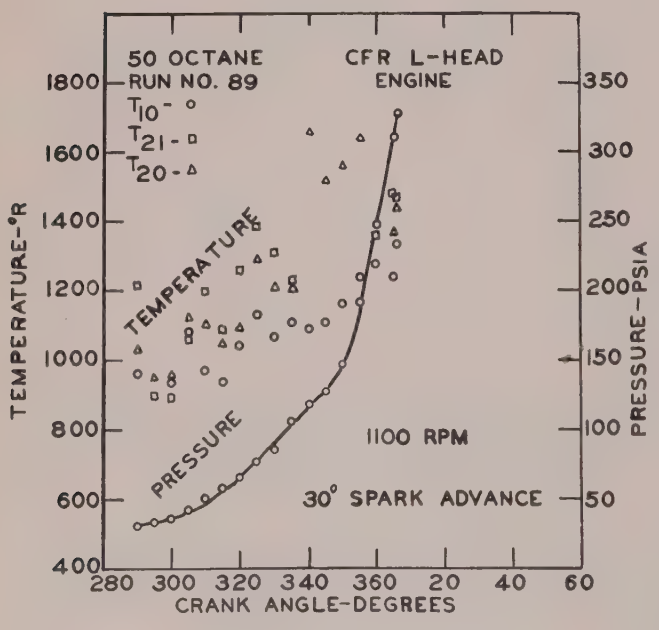

Fig. 10 – Data showing extreme scatter in temperature values

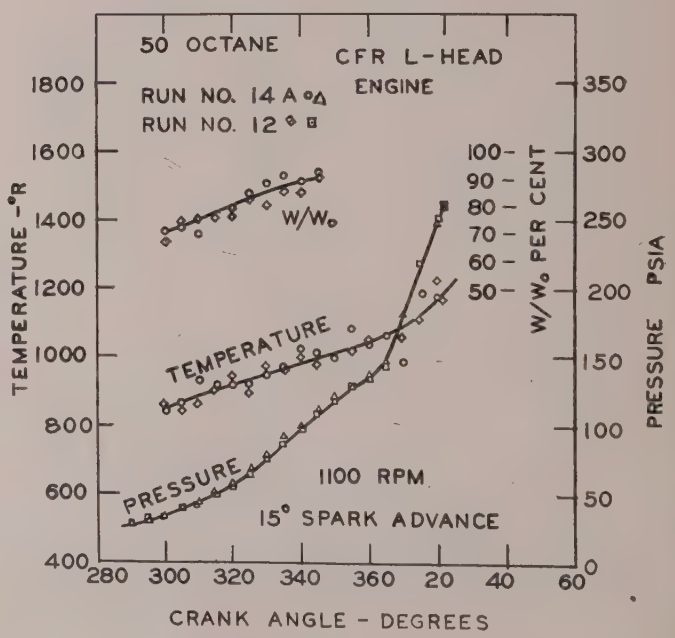

Fig. 11 – Data showing reproducibility of measured temperatures and pressures using 50-octane fuel

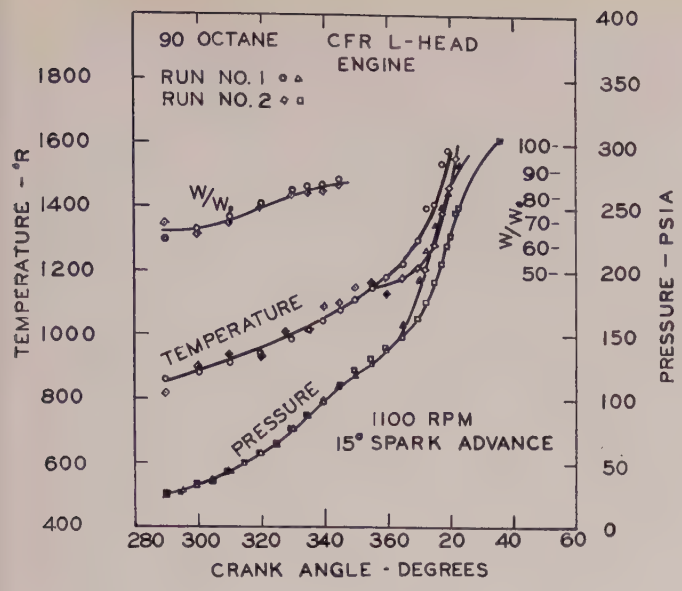

Fig. 12 – Data showing reproducibility of measured temperatures and pressures using 90-octane fuel

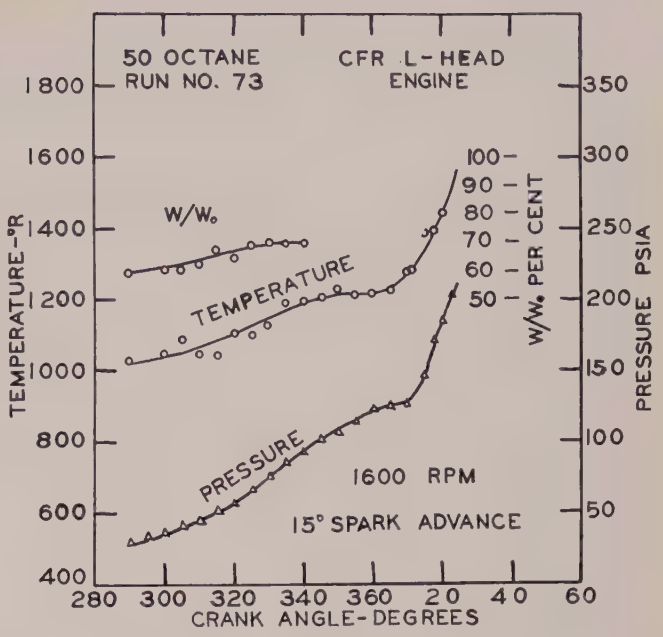

Fig. 13 – Temperature-pressure data taken at 1600 rpm using 50-octane fuel

bined into a single curve if the data were plotted as temperature versus pressure. This resulting single curve would be expected if the exponent n in the expression PV^n did not change with spark advance and if the weight of charge in the cylinder did not vary with spark advance. The trial plots showed n and the charge weight to be constant within the accuracy of the experimental data. Accordingly, the data for the three fuels were so plotted and combined together as illustrated in Fig. 15. Logarithmic scales were chosen to show, qualitatively, the effect of heat transfer and heat released in the end gas. Fig. 15 shows data taken at 1100 rpm, while Fig. 16 shows data taken at 1600 rpm.

Discussion of Results

The validity of the temperatures determined from the iodine absorption spectrum requires that there be no absorption by gases other than iodine. Comparison of the film records shown in Figs. 6 and 7 shows that for the data without iodine there was negligible absorption during the compression stroke. The data with iodine show absorption

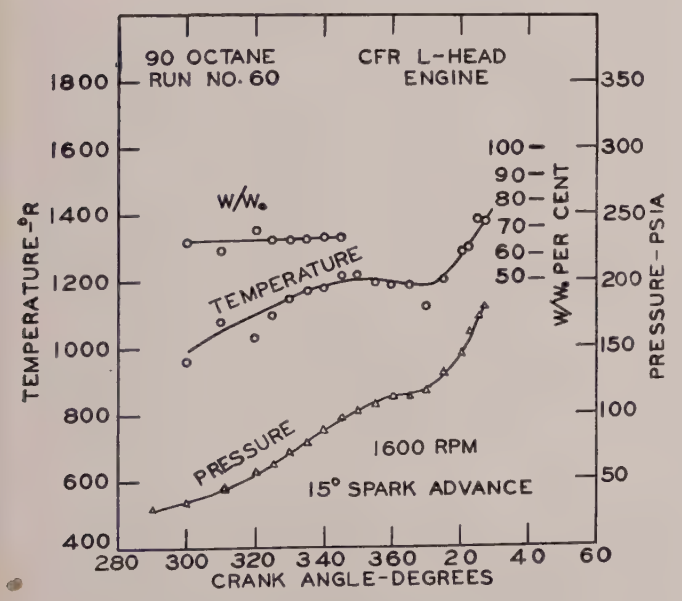

Fig. 14 – Temperature-pressure data taken at 1600 rpm using 90-octane fuel

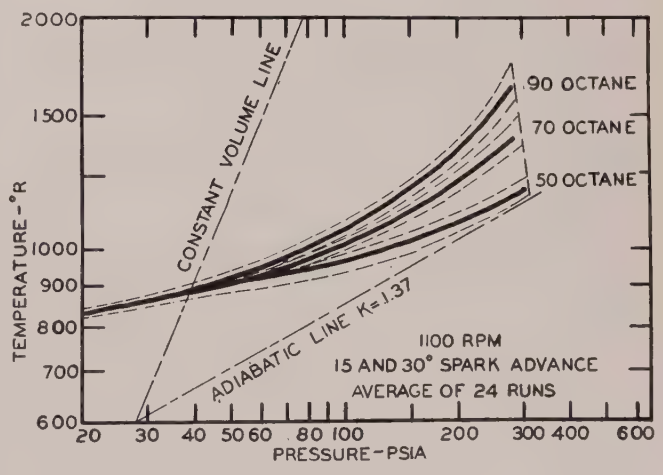

Fig. 15 – Logarithmic plot of pressure versus temperature for 1100-rpm data

(downward deflection of the trace indicates absorption, the flat portion of the trace at about 220-deg crank angle corresponds to zero absorption and was produced by a light chopper) at three wavelengths, I_0, I_1, and I_2, until about 20 deg after top center. The sharp decrease in the absorption signals at this point was attributed to the effect of the flame front reaching the light path and dissociating or consuming the iodine. This effect of the flame was deduced from the fact that the decrease of the iodine absorption signal occurred at approximately the same point in the cycle that radiation from the flame was detected when the light source was turned off. Dependable end-gas temperature measurements could not be made beyond the point at which the flame front reached the light path.

The typical scatter in temperature data as determined from three different wavelengths (a combination of any two absorption readings provides one temperature determination) is shown in Fig. 8. The scatter was attributed to various factors, such as the inherent noise level in the signal, as evidenced by the small fluctuations in the absorption traces in Fig. 6, inaccuracies in the zero absorption readings, and the ever-present scaling errors. Rapid changes in window transmissivity, selective absorption by gases other than iodine, phototube drift, and calibration errors may have contributed to the scatter. However, these latter factors are considered to have a relatively small effect as compared to signal-to-noise ratio and changes in the zero absorption reading, except under the conditions mentioned in connection with Fig. 10, where rapid changes in window transmissivity are suspected.

Comparison of the data shown in Figs. 11 and 12 with the data shown in Figs. 13 and 14 indicates that higher compression temperatures are obtained at the higher engine speeds. This is qualitatively what would be expected, since less heat per cycle is lost to the cylinder walls at higher engine speeds. Also, the quantity W/W_0 is about 10% lower at 1600 rpm than at 1100 rpm. This decrease is reasonable both in magnitude and in direction, since W/W_0 is analogous to volumetric efficiency and in general the volumetric efficiency of a naturally aspirated 4-stroke engine decreases with increasing speed. These results indicate that the iodine absorption technique was providing reasonable values for end-gas temperatures. In this connection experience had shown that the zero absorption reference line on the exhaust stroke gave much more reliable and consistent readings than the zero reference line obtained from the synchronized light chopper. Thus, it was impractical to obtain data on an unfired engine, since no zero absorption reference line such as is obtained during the exhaust stroke on a fired engine was available.

In relation to data taken under engine knocking conditions, it was noted that for all the data obtained, even under the most severe knocking conditions, the flame front entered the light path at least 8 to 10 crank angle deg prior to knock. It is unfortunate that the light path arrangement did not permit obtaining continuous temperature measurements up to the point at which knock occurred. However, a study of published pictures of flame fronts in a similar engine[6] and an analysis of flame front data by Melby et al[7] taken by means of ionization gaps indicate that the end gas might be located either over the exhaust valve or above the piston. In future studies it is planned to vary the spark plug, the quartz window location, and possibly to modify the combustion-chamber design slightly in an attempt to ensure that the end gas is located in front of the quartz windows. For a given fuel no marked differences were observed between compression temperatures for knocking and nonknocking combustion.

Inasmuch as the temperatures presented are experimentally determined vibrational temperatures of iodine, the question arises as to the effect of iodine on the precombustion reactions and also the effect of the precombustion reactions on the iodine molecule. There is some indication in the published literature that iodine, as an additive, inhibits knock, although the results are not conclusive.[8] Because of this uncertainty experimental data were obtained by both Ethyl and du Pont. A personal communication from H. E. Hesselberg of Ethyl gave the following information:

1. In a blend of 60% isooctane, 40% n-heptane, 0.40% by weight elemental iodine (10.33 g of iodine

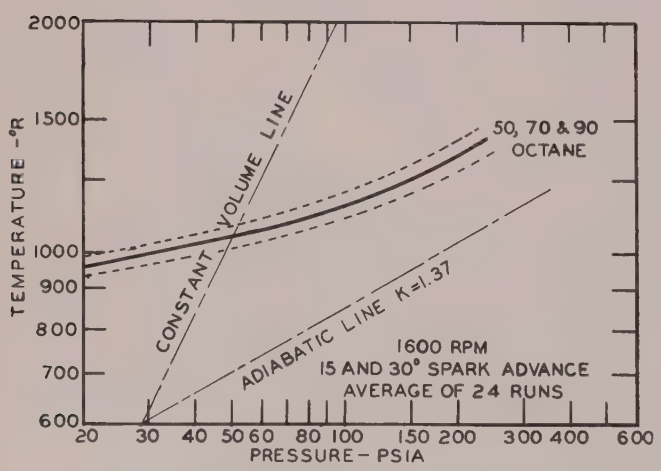

Fig. 16 – Logarithmic plot of pressure versus temperature for 1600-rpm data

[6] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-752: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposit," by L. L. Withrow and F. W. Bowditch.

[7] See "An Investigation of Preignition in Engines," by A. O. Melby, D. R. Diggs, and B. M. Sturgis, on pp. 32-39 of this volume.

[8] See "Explosion and Combustion Processes in Gases," by W. Jost. Pub. by McGraw-Hill, New York, 1946.

per gal) gives an approximately saturated solution.

2. By both Motor and Research Methods at concentrations of from 0.10 to 0.40% by weight, the elemental iodine exhibited definite antiknock properties with a maximum increase in antiknock level of about four octane numbers.

3. In this isooctane-n-heptane blend the elemental iodine was 1-2% as effective as lead introduced as tetraethyl lead on a weight basis.

Tests conducted at the du Pont Petroleum Research Laboratories indicate that the addition of 10 g iodine per gal of normal heptane had a negligible effect on the autoignition and cool-flame limits of the air-fuel mixture (Fig. 17). They also measured the increase in octane rating for a 60-octane primary reference fuel containing 10 g iodine per gal and found an increase of 2.5 octane numbers.

Although the quantity of iodine used for temperature measurement was only about 0.02% of the mixture volume, this amounted to a weight ratio of approximately 2% iodine to fuel. It was observed that the addition of this quantity of iodine reduced the knocking tendency, although exact quantitative measurements were not made. Variation of iodine concentration from 0.01 to 0.04% by volume seemed to have no marked effect on end-gas temperatures. It was also noted that the indicated concentration of iodine per unit weight of end gas varied only about ±10% from 60 deg before top center to 20 deg after top center. This should indicate that little, if any, iodine was combining with the fuel or air or products thereof, since only uncombined iodine gas will indicate absorption (and thereby concentration) in the iodine absorption spectrum.

There were no indications that iodine had an appreciable effect on preflame reactions. Furthermore, experimental evidence such as the comparative constancy of the ratio W/W_0 and the increase in temperature with speed followed expected trends. However, it must be borne in mind that the temperatures measured in these tests corresponded to the vibrational temperature of iodine molecules and conceivably may not have indicated true equilibrium temperatures. Beck et al[3] discuss the possible ways in which the measured temperatures might be affected by the precombustion reactions and, although no experimental proofs are available, it is the general conclusion that the precombustion reactions are not likely to affect the temperature measurements. This reference also discusses the possible effect of pressure on the measured temperatures and again, while no experimental proofs are offered, it is concluded that pressure has no effect other than that expressed by $PV = RT$. It should be noted, however, that any temperature reading obtained from a system that is not in temperature equilibrium will depend upon the measuring technique. Thus, if the temperatures measured by the iodine absorption technique are to be compared with those obtained by another technique ($PV = WRT$, for example) the assumption of temperature equilibrium in the system must be made.

The curves in Figs. 16 and 17 were plotted on a logarithmic scale to show qualitatively the heat released in the end gas. Assuming temperature and pressure equilibrium throughout the charge the resulting curves show the combined effects of heats of reaction (both positive and negative), heat loss to the cylinder walls, the energy added by work done during compression, and the change in the number of moles. If the number of moles is constant, any slope greater than the slope for an isentropic compression indicates a net effect equivalent to an addition of heat. If it is assumed that the measured vibrational temperature of the iodine gas corresponds to the equilibrium end-gas temperature, it can be deduced that at 1100 rpm the heat released in the end gas is greater for 90-octane than for 50-octane fuel for the particular operating conditions, or that the change in the number of moles is sufficient to account for the differences shown.

The results presented may at first seem to be in disagreement with the results of other investigators of precombustion reactions.[9, 10, 11] However, it must be noted that these investigations involved determinations of heats of reaction from pressure measurements alone and were obtained at lower pressure levels and under somewhat less severe

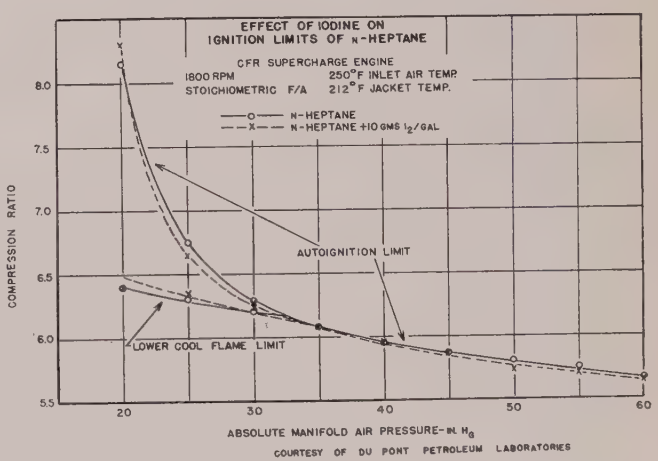

Fig. 17 – Autoignition and cool-flame limits with iodine in engine cylinder

[9] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 571-587: "Precombustion Reactions in a Motored Engine," by D. L. Pastell.

[10] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 488-508: "Some Effects of Fuel Structure, Tetraethyl Lead, and Engine Deposits on Precombustion Reactions in Firing Engine," by W. Cornelius and J. D. Caplan.

[11] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 472-487: "Early Combustion Reactions in Engine Operation," by E. B. Rifkin, C. Walcutt, and G. W. Betker, Jr.

operating conditions than the data presented here.

More recent results[5] of end-gas temperature measurements using the iodine absorption technique emphasize that a 2-stage precombustion reaction may occur, and offer some evidence of a negative temperature coefficient.[12] There is indication that the first-stage reaction is completed shortly before top center and the second-stage reaction starts near top center. It is entirely possible that fuel of lower octane produces greater heat release during the first-stage reaction than does fuel of higher octane number, and that the reverse is true for the second-stage reaction. There is further evidence in the literature of 2-stage combustion of hydrocarbons.[13, 14, 15] Lewis and von Elbe[14] maintain that the end of the first-stage reaction is accompanied by cool flames. Taylor, et al[15] have observed 2-stage ignition in both n-heptane and isooctane.

The possibility of a 2-stage preflame reaction suggests that the results of Cornelius and Caplan[10] and Rifkin, et al[11] correspond to the first-stage reaction. This can account for the reversal in the trend of heat released with the octane rating of the fuel.

According to the nuclear ignition theory as proposed by King,[16] the preflame reactions of hydrocarbons in an engine are a race between cracking and oxidation reactions. The cracking process, according to King, produces carbon particles, which promote autoignition and knock. The oxidation reactions, on the other hand, produce CO, CO_2, and H_2O, all of which tend to inhibit autoignition in an engine. If the oxidation reactions are assumed to be more exothermic than cracking reactions the results shown in Fig. 9 can be considered in agreement with the portion of King's theory dealing with the cracking and oxidation reactions.

Conclusions

From the results of this investigation the following conclusions were drawn:

1. The temperature of iodine gas can be determined from its light-absorbing properties.

2. The light-absorbing properties of iodine gas can be measured instantaneously and continuously in an operating spark-ignition engine at gas temperatures below 1500 F.

3. The vibrational temperature of iodine molecules, in a firing engine, can be determined with an estimated accuracy of ±50 F.

4. Further studies should be made to:

a. Improve the experimental technique.

b. Determine the effect of iodine on combustion, and vice versa.

c. Determine experimentally if pressure has an effect other than that expressed by $PV = WRT$.

d. Check the results by a different temperature-measuring technique.

5. If light absorption by iodine correctly indicates gas temperatures (and all indications are that it does) there is a considerable difference between high- and low-octane primary reference fuels in type and effect of the preflame reactions occurring for certain operating conditions.

All indications were that the method described correctly measures the temperature of the unburned gases in a spark-ignition engine. If further investigations continue to verify these results, the technique should prove to be a valuable tool for investigating engine combustion phenomena.

Acknowledgment

Acknowledgment is due to the Coordinating Research Council, to the General Motors Corp., to the Wisconsin Alumni Research Foundation, to the College of Engineering, and to the National Advisory Committee for Aeronautics for financial and equipment aid. Especial acknowledgment is due J. G. Winans, of the Physics Department, for his assistance in developing the absorption technique.

APPENDIX I

When radiation passes through a uniform absorbing gas the intensity is reduced in accordance with the Beer-Lambert law by:

$$\text{Log} \frac{I_{a_\lambda}}{I_{o_\lambda}} = -K_\lambda d \qquad (1)$$

where I_{a_λ} and I_{o_λ} are the radiant power per unit wavelength range and per unit cross-sectional area at wavelength λ, which enter and leave the absorbing region, respectively; d is the length of the light path; and K_λ is the absorption coefficient at wavelength λ. $K_\lambda d$ is proportional to the number of molecules N_λ in the light path, which have initial energy states that permit absorption at wavelength λ or:

$$K_\lambda d = \alpha_\lambda N_\lambda \qquad (2)$$

[12] See *Journal of American Chemical Society*, Vol. 60, September, 1938, pp. 2244-2246: "Negative Temperature Coefficient in Rate of Propane Oxidation," by R. M. Pease.

[13] "Autoignition in an Internal-Combustion Engine," by W. J. Levedahl and C. W. Sargent. B. S. thesis, M.I.T., 1948.

[14] See "Combustion, Flames, and Explosions of Gases," by B. Lewis and G. von Elbe. Pub. by Academic Press, New York, 1951.

[15] See SAE *Quarterly* Transactions, Vol. 4, April, 1950, pp. 232-274: "Ignition of Fuels by Rapid Compression," by C. F. Taylor, E. S. Taylor, J. C. Livengood, W. A. Russell, and W. A. Leary.

[16] See *Canadian Journal of Research*, Vol. 26, Sec. F, May, 1948, pp. 228-240: "Oxidation, Ignition, and Detonation of Fuel Vapors and Gases, Part IV—Cause of Detonation or Combustion Knock in Engines," by R. O. King.

where α_λ is a molecular absorption coefficient.

At any given wavelength, the absorption could consist of lines in the P, O, and R branches of several overlapping bands. Thus:

$$N_\lambda = f_{ip_\lambda} N_o + f_{iq_\lambda} N_o + f_{ir_\lambda} N_o \cdots f_{np_\lambda} N_n + f_{nq_\lambda} N_n + f_{nr_\lambda} N_n \quad (3)$$

where f_{n_λ} is the fraction of molecules of vibrational energy E_n that can absorb radiation of wavelength λ. For example, f_{nq_λ} is the fraction of the molecules in the vibrational energy state n and with rotational energy such that absorption of radiation at the wavelength λ corresponds to one line in the Q branch.

The Maxwell distribution law for molecular rotation gives:

$$f_{nq_\lambda} = \frac{hCB}{kT}\left(2J_{nq_\lambda} + 1\right)_e - \frac{hCB}{kT} J_{nq_\lambda}\left(J_{nq_\lambda} + 1\right) \quad (4)$$

where h is Planck's constant, C is the velocity of light, B is the molecular rotation constant, k is Boltzmann's constant, and T is the temperature. Similar expressions can be written for f_{np_λ} and f_{nR_λ}.

In order to obtain a relationship between temperature and the amount of light absorbed, it is necessary to combine equations (1), (2), (3), and (4) and to integrate the resulting equation over the range of wavelengths admitted to the phototube. Unfortunately, the resulting equation is of sufficient complexity that it cannot be integrated mathematically without making numerous approximations of questionable validity. The equation obtained, however, is:

$$\text{Log}\ \frac{\log \frac{D_a}{D_o}}{\log \frac{D_b}{D_o}} = A + \frac{\overline{E_b} - \overline{E_a}}{kT} \quad (5)$$

where D_a and D_b represent the phototube readings where absorption occurs over ranges of wavelengths, D_o represents the photo reading with no absorption, A is a constant, and $\overline{E}_a$ and $\overline{E}_b$ are the known average values of vibrational and rotational energy. The experimental calibration data obtained with the setup shown in Fig. 3 are approximately represented by equation (5) but, since the maximum of accuracy was desired, the experimentally obtained curve was used.

One other item of interest is the effect of pressures and chemical reactions on the iodine spectrum, particularly if they caused the experimentally determined vibrational temperature to differ from the rotational and vibrational temperature.

The effect of a change of pressure on an absorption spectrum is a change in width of absorption lines without any change in the total intensity of absorption from a continuous spectrum source. The iodine absorption bands consist of many closely spaced lines. An increase in pressure causes each line to broaden and overlap neighboring lines to a greater extent than before. If there is no discontinuity in the spectrum, as much new radiant power should be brought in at the edges of the slit as is removed. Thus, a change in pressure should have no effect on the phototube readings.

With regard to the effects of chemical reactions, the measurement made is the absorption due to I_2 molecules. Thus, the only chemical reactions of interest are those affecting or producing I_2 molecules. Those reactions that simply destroy I_2 molecules are not of interest, since the temperatures obtained are independent of the concentration of the I_2 molecules.

Chemical reactions might produce molecules with abnormally high vibrational energies, and these in turn might conceivably produce iodine molecules with abnormally high vibrational temperatures. However, the concentration of the I_2 molecules is low and the chances of this happening are small. It is also true that, if an I_2 molecule of abnormally high vibrational energy was produced, it would – because of its high energy state – absorb at wavelengths longer than those used in the present measurements and have no direct effect on the measured absorption. Thus, it is difficult to see how the measured vibrational temperatures could differ from the rotational and translational temperatures due to any anomalous effects of pressure or chemical reactions. It is true, however, that the measured temperature is a spatially averaged temperature where any temperature gradients exist in the light path.

The correctness of the above viewpoint is borne out by the statement made by Gaydon,[17] "with low-pressure flames of acetylene burning with oxygen, it seems that in the reaction zone the rotational temperature of the OH radicals as determined from a study of the 3064 A band in emission is much too high, being around 5000 to 6000 C. Above the reaction zone the rotational temperature is more of the right order of magnitude. The absorption bands of OH in the same flame appear to give a reasonable value for the temperature, even in the reaction zone." Since, in these measurements only absorption bands were used, it can be expected that the temperatures obtained are correct within the error of measurement.

Discussion of this paper – as well as of the Millar, Uyehara, Myers paper, which follows – is to be found on pp. 525-530.

[17] "Spectroscopy and Combustion Theory," by A. G. Gaydon. Pub. by Chapman & Hall, London, 1948.

Practical Application of Engine Flame

BOTH temperature and pressure are customarily considered as fundamental thermodynamic variables in an engine. It is logical, therefore, that the measurement of these variables in an operating engine has attracted the attention of many workers. Although reliable pressure measurements are not yet simply made, the state of the art is considerably more advanced than that of temperature measurements. Thus, it is understandable that wide variations in both the technique and the results of temperature measurements are found (Fig. 1).

The wide variations in the measured temperature as shown in Fig. 1 are caused in part, at least, by the fact that large temperature inhomogeneities exist in the combustion chamber even during normal combustion where it is generally thought that the pressure inhomogeneities are small. In the absence of knock and extensive preflame reactions, the combustion cycle may be divided into three parts as illustrated in Fig. 2. In the first phase, that is, from the time the intake valve is closed until spark occurs, the temperature inhomogeneities would seem small, although some temperature gradients caused by heat transfer, hot clearance gases, and precombustion reactions occurring before spark might be present. During this phase it would be logical to expect that the measured temperatures would correlate reasonably well with the relationship $PV = WRT$. In the second phase, which is considered to be from the time of spark until combustion is completed, large temperature inhomogeneities are present even during normal combustion with the comparatively cold unburned gases separated from the hot burned gases by the flame front. In addition to the large temperature inhomogeneities just mentioned, any inhomogeneities existing in phase one may be continued and perhaps accentuated. In addition, heat transfer and the Hopkinson effect[1] may introduce new inhomogeneities. Thus, during phase two, measurements of the temperature of either the burned or the unburned gases might bear some relationship to $PV = WRT$ (provided the respective volumes were known), but measurement of the average temperature may bear little, if any, correlation with the cylinder volume and pressure. Thus, during this

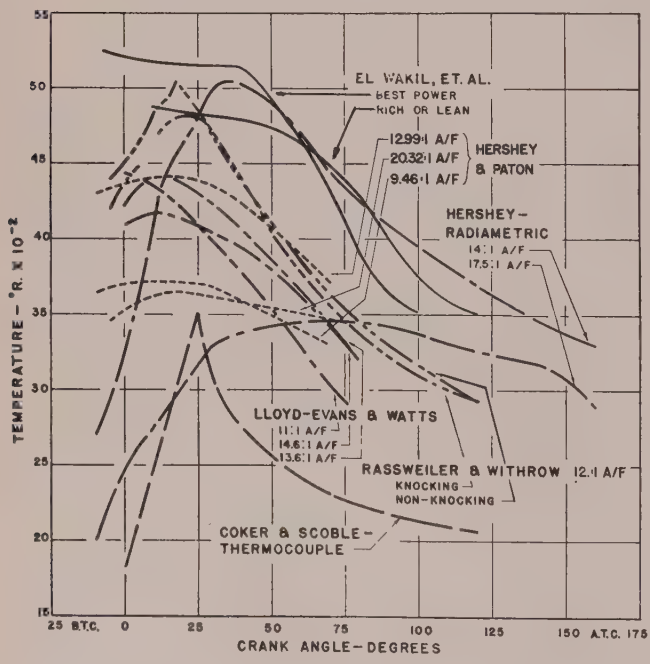

Fig. 1 – Temperature measurements made by different investigators

[1] See Proceedings of Royal Society of London, Vol. 77, Series A, June, 1906, pp. 387-413: "Explosions of Coal Gas and Air," by B. Hopkinson.

Temperature Measurements

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This paper was presented at the SAE National Fuels & Lubricants Meeting, Chicago, Nov. 6, 1953.

phase the temperature values are often meaningless, unless the averaging technique is specified.

When combustion is complete (phase three), some temperature inhomogeneities due to heat transfer and the Hopkinson effect may still exist; but a reasonably good correlation of measured temperature with the relationship $PV = WRT$ could be expected.

The temperature data reported in this paper were taken during the second and third phases and are believed to represent a reasonable value for the temperature of the burned gases.

Engine and Engine Instrumentation

The engine used for these investigations was a single-cylinder spark-ignited CFR engine. Modifications were made to the engine cylinder head to provide a continuous light path through the combustion chamber. A cross-section of this cylinder head is shown in Fig. 3. A total of seven holes were available for insertion of different measuring devices.

Airflow to the engine was measured by a critical-flow technique.[2] In the critical-flow technique, the absolute upstream pressure and temperature of the air is used as a measure of the weight of air flow-

[2] "Effect of Selected Ignition Accelerators on Diesel Combustion Temperatures," by W. Chow, Ph.D. Thesis, University of Wisconsin, 1952.

THIS paper presents flame temperature and pressure data taken on a spark-ignited CFR engine. Data are presented for the four variables of cyclic reproducibility, knock, air/fuel ratio, and ignition timing.

The data indicate that cycle-to-cycle irreproducibility may be caused by variations in the initial rate of flame propagation from the spark.

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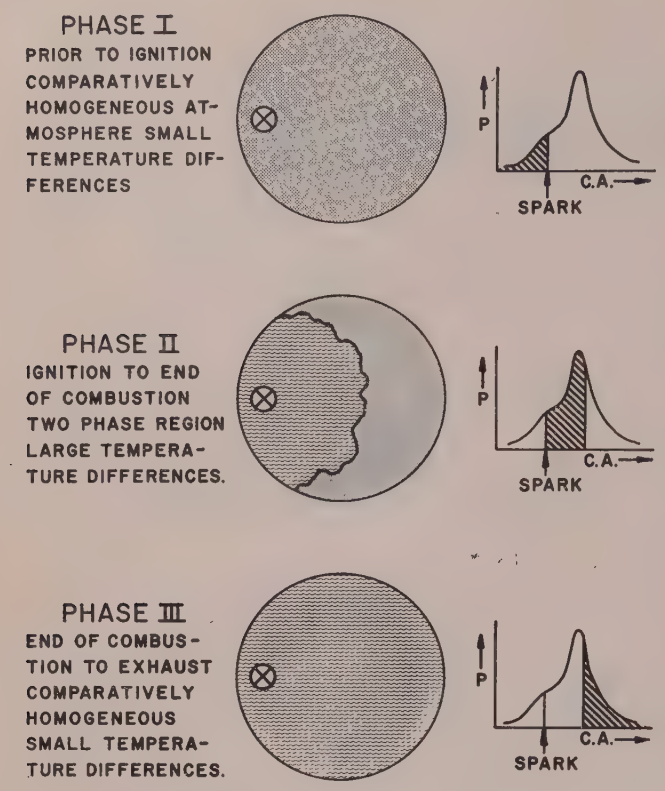


Fig. 2 – The three phases of combustion in a spark-ignited engine

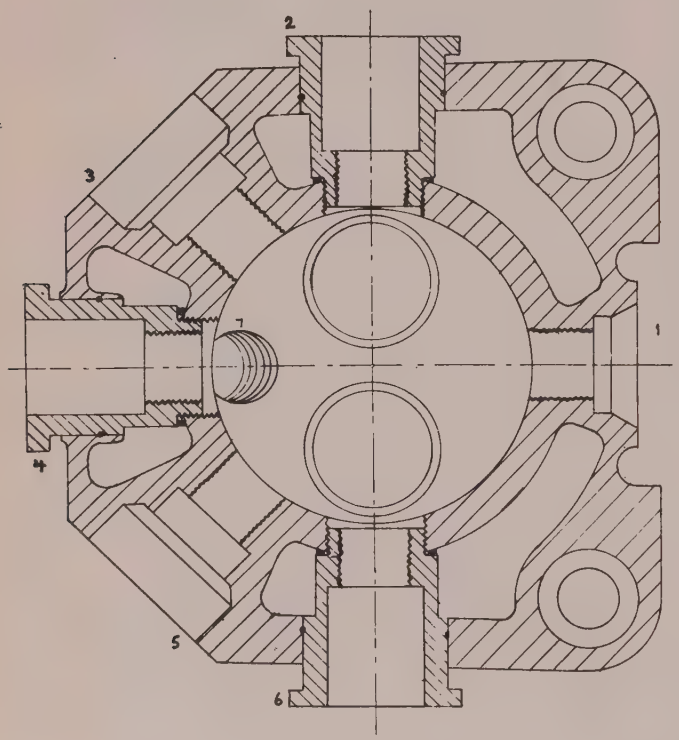


Fig. 3 – Cross-section of cylinder head

ing. The air was assumed to be saturated with water vapor at the temperature and pressure in the line coming into the test cell. This seemed reasonable in view of the condensate visible from other air outlets. Manifold pressure was controlled by a diaphragm-operated valve and a pneumatic controller which maintained a constant manifold pressure at any preset value through the range available.

The fuel flow to the engine was measured by means of an automatic weighing system which would time the flow of 2 oz. of fuel. Reproducibility of the fuel-measuring equipment was within 1%.

Since continuous temperature measurements were to be made, one of the prerequisites of cylinder-pressure measurement was that the device or instrument used must be continuous over any given engine cycle. The use of a strain-gage-type Li indicator with continuous calibration of both the indicator and the electronic circuits was felt to be sufficiently accurate for this work. Since it was desired to record the cylinder pressure at every instant of crank-angle position and at the same time superimpose calibration marks on the pressure trace, a balanced-diaphragm pressure indicator was used to mark the pressure record with a known pressure calibration on every cycle. To establish the scale factor for reproducing a pressure-time record, two such calibration marks were necessary. The second mark was taken at atmospheric pressure.

In order to establish a calibration mark at atmospheric pressure, the balanced-diaphragm pressure indicator was opened to the atmosphere. This technique allowed a calibration mark to be placed on the pressure record indicating atmospheric pressure. Inasmuch as only one balanced-diaphragm pressure indicator was available, it was impossible to place the high-pressure and atmospheric calibration on the pressure-time record simultaneously. As a result the crank-angle position at which atmospheric pressure occurred was observed over a number of engine cycles. It was noted that this atmospheric marker appeared consistently at 210 deg. atdc during the exhaust stroke for all conditions of operation for which data are presented. This crank-angle position was identified with the atmospheric calibration in all future records. All records carried the high-pressure calibration mark.

The value of the calibration pressure applied to the balanced diaphragm was adjusted to a pressure slightly under the peak pressure of a given engine cycle. In this way maximum calibration accuracy was obtained.

The important features of the pressure-measuring technique are as follows:

1. Continuous pressure measurement over each engine cycle.

2. Calibration superimposed on each pressure record.

3. Simple and positive recording.

Fig. 4 shows the static calibration of the indicator and associated electronic equipment. For all practical purposes the indicator was linear and seemed insensitive to the temperature changes encountered.

The temperature of the burned gases was measured by a modified sodium-line-reversal technique.[3, 4] Briefly, the technique is as follows: Two characteristics describe the radiation from a flame for a given wavelength range, that is, its temperature and its emissivity. Since absorptivity equals emissivity, measurements of absorptivity and radiation intensity from a flame in a narrow wavelength range permit the determination of the temperature of that portion of the flame in the light path. In the arrangement used for this work, "chopped" light from a molten zirconium-arc lamp passed through the combustion chamber onto the entrance slit of a spectrograph. This varying radiation intensity was measured by a photomultiplier located at the sodium wavelength. The decrease in intensity of the "chopped" light was a measure of the absorptivity of the flame. When the light was interrupted by the chopper, the same photomultiplier measured the radiation from the flame alone. Combination of the two readings enabled the temperature of the flame to be determined. The instrument is simple in theory and operation and seems to give reliable and reasonable results.

In order to use the sodium-line technique to measure the temperature of the flames in the combustion process, the flames must be colored with sodium. This was done by introducing sodium into the fuel. A very small amount of a sodium solution was enough to produce adequate coloring of the flame. In order to introduce the sodium into the fuel, sodium methoxide was dissolved in methyl alcohol to form a solution. This solution was mixed with equal parts of acetone. This mixture was then mixed 10% by volume with the test fuel. After thorough mixing, the fuel was allowed to stand and the undissolved sodium mixture decanted. The removed portion amounted to about 9%. As a result, only a small amount of the solution actually remained in the fuel – about 1%.

The accurate determination of cylinder volume at any instant in the engine cycle is as equally important as pressure and temperature measurement. A pickup was designed to indicate at intervals of every crank-angle degree the position of the engine crankshaft and thus the volume of the cylinder. It should be noted that torsional vibration and deflection of the crankshaft and engine parts may alter this relationship. However, their effect was believed to be small. The instrument used for this purpose is a magnetic pickup or transducer located very close to the periphery of the engine flywheel. As the engraved 1-deg crank-angle timing marks pass this pickup a voltage is induced proportional to the rate at which the lines of flux are cut by the engraved timing marks. This signal is shaped by a pulse-shaping circuit and applied to the vertical deflection plates of an oscilloscope. A resulting mark appears for every degree of crank-angle rotation.

To record photographically engine data and to provide a time axis, a drum camera was used to photograph the oscilloscope traces. Figs. 5 and 6 are typical drum-camera records. Fig. 5 shows portions of the drum-camera record as obtained from the drum camera. Fig. 6 illustrates how the film was marked for analysis purposes. It should be noted that two consecutive cycles are shown – the first fired, and the second unfired. The unfired cycle is shown for comparative purposes and was obtained by interrupting the spark for a single cycle. Starting with the top trace of Fig. 5, the traces are as follow:

Trace 1 – This is the pressure trace for a fired

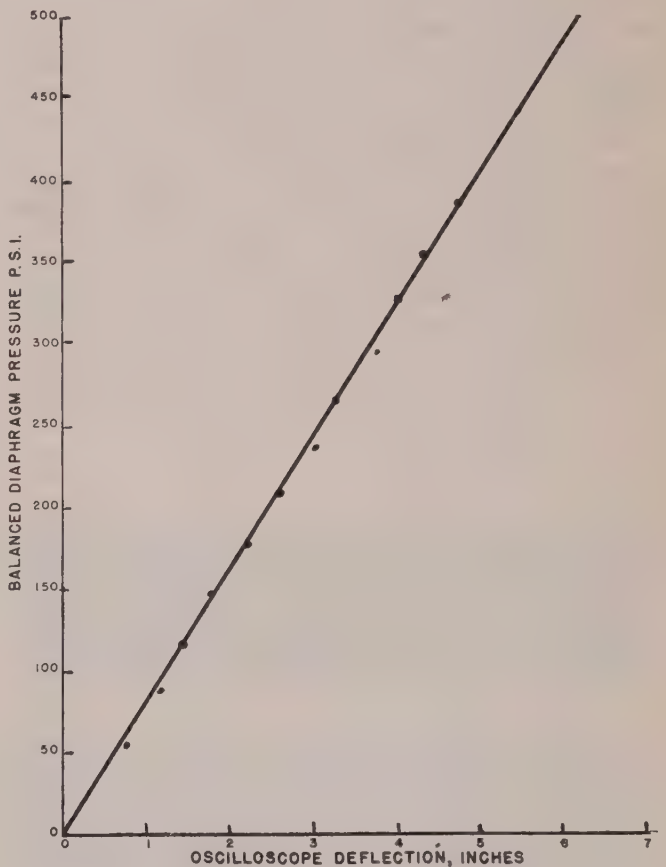


Fig. 4 – Static calibration of pressure indicator

[3] See *Journal of Optical Society of America*, Vol. 43, July, 1953, pp. 609-617: "Fast, Electro-Optical, Hot-Gas Pyrometer," by G. H. Millar, J. G. Winans, O. A. Uyehara, and P. S. Myers.

[4] "Flame Temperature Measurements as Aid to Internal-Combustion-Engine Analysis," by G. H. Millar, Ph.D. Thesis, University of Wisconsin, 1952.

cycle. A calibration mark corresponding to a known pressure may be seen on the pressure trace.

Trace 2 – This is the motoring pressure trace of the unfired cycle following the fired cycle.

Trace 3 – This is an alignment trace produced by a small lamp operated from direct current and located behind a small hole on the front of the oscilloscope. All measurements were made from this line.

Trace 4 – These are the 1-deg timing marks associated with the fired trace. They are generated by the small magnetic pickup located close to the flywheel of the engine as previously described. The double mark close to the left edge of the film identifies 90 deg btdc.

Trace 5 – These timing marks are associated with the unfired cycle, Trace 2. They are generated by the same device as those of Trace 4 but are automatically lowered on the oscilloscope screen by a relay, cam, and microswitch arrangement.

Trace 6 – This is the ignition trace. The spark occurs at the first interruption of the trace when reading from left to right.

Trace 7 – This is the flame-temperature trace obtained as previously explained.

Trace 8 – This is another alignment line similar to Trace 3.

Trace 9 – This trace cannot be seen on the record as photographic definition has been lost in reproduction. It consists, however, of a high-gain channel which amplifies the bottom envelope of Trace 7. In this manner, additional temperature readings may be made when the scaling error on Trace 7 becomes large because of the small deflections.

Actually two more oscillograph traces than those shown in Fig. 5 were available. The two traces not shown were used for 10-deg timing marks and rate-of-change-of-pressure measurements. Neither of these has proved useful as yet on recorded data, but on occasion they have proved useful for visual observation.

In order to scale accurately the amplitudes of the oscilloscope trace recorded on the film, an optical comparator was used which employed an optical magnification of 20 diameters. The film was first marked as shown in Fig. 6 and then inserted into the film holder of the comparator. An image of the film was projected on a ground-glass screen containing a set of cross-hairs. It was possible to make measurements in the vertical direction to 0.0001 in. and measurements in the horizontal direction to 0.001 in. by means of micrometers located on the film carrier. The practical limit to the scaling accuracy was the width and fuzziness of the lines on the film, since the average line width ran from 0.003 to 0.006 in. All measurements were made from alignment lines in order to minimize any film warpage or misalignment. All data were photographed on linograph-pan film.

Table 1 – Engine Operating Conditions

	Knock	Spark Advance	Air/Fuel Ratio
Air/Fuel Ratio	11/1	11/1	Variable
Jacket Water Temperature, F	212 – 10	212 – 10	212 – 10
Spark Advance, btdc	35 ± ½	Variable	35 ± ½
Compression Ratio	6/1	6/1	6/1
Rpm	1250 ± 50	1250 ± 50	1250 ± 50
Manifold Pressure, in. of Hg	29.5 ± 0.1	30.5 ± 0.1	30.5
Crankcase-Oil Temperature, F	155 ± 5	155 ± 5	155 ± 5
Mixture Temperature, F	54 ± 1	54 ± 1	54 ± 1
Fuel Used:			
Knocking	60% isooctane 40% *n*-heptane	Isooctane	Isooctane
Nonknocking	Isooctane		

Procedure and Data

The general procedure for recording data was as follows: The engine was allowed to operate until the jacket water temperature had reached 212 F. This usually required about 1 hr. During this time, the quartz windows were replaced by metal plugs of the same shape and general character as the quartz windows.

When the engine reached equilibrium operating conditions, the air and fuel flow were measured and adjusted until the desired air/fuel ratio had been reached. Both air and fuel flow measurements

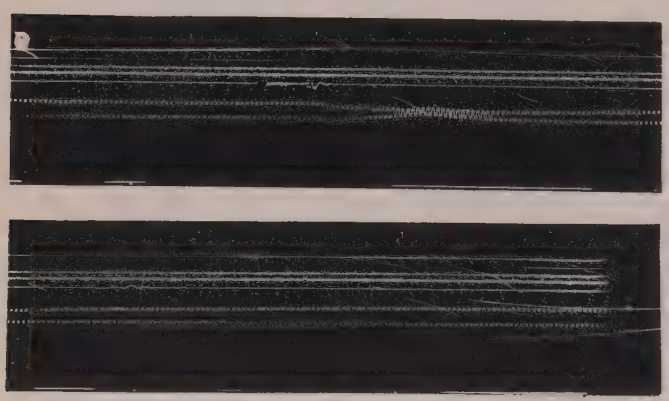


Fig. 5 – Portion of typical film from drum camera – before marking

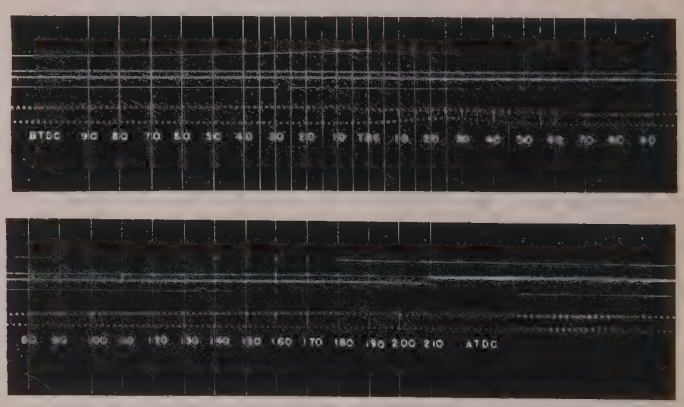


Fig. 6 – Portion of typical film from drum camera as marked for analysis purposes

were expressed on a weight basis. Although it was originally intended to study only the three variables of knock, air/fuel ratio, and spark advance, it was found early in the study that cycle-to-cycle variations were large, and consequently a study was made of cyclic reproducibility. The engine operating conditions which were maintained for the different tests are shown in Table 1.

The extent of the cycle-to-cycle variations are shown in Fig. 7 where the measured pressure, temperature, and flame emissivity are plotted as a function of crank-angle for four different cycles. Also shown in Fig. 7 is the weight of charge inducted as determined from fuel and air measurements and the calculated weight of charge. The calculated charge weight was determined from $PV = WRT$ where the variables were determined as follows:

P = Pressure, measured.

V = Volume, total cylinder volume at crank angle in question. The use of this volume in the equation in conjunction with the temperature of the burned charge explains why the calculated weight increases to and reaches a constant value.

W = Weight, calculated.

R = Gas constant, determined from Hershey, Hottel, and Eberhardt combustion charts[5] at the pressure, temperature, and air/fuel ratio in question.

T = Temperature, measured.

Fig. 8 presents similar data for four knocking cycles and serves not only to illustrate cycle-to-cycle reproducibility but to illustrate the effects of knock as well.

In presenting the data for the variables of spark-advance and air/fuel ratio, a number of individual cycles of a progressively changing variable are shown. The results are shown in Figs. 9, 10, 11, and 12 for the case when the variable was ignition timing, and in Figs. 13, 14, 15, and 16 for the case when the variable was air/fuel ratio.

Discussion of Results

Unfortunately, it seems impossible except by inference to determine the correctness of the measured temperature. The fact that the calculated weights level off to a constant value is encouraging,

[5] See SAE Transactions, Vol. 31, (October) 1936, pp. 409-424: "Thermodynamic Properties of Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel.

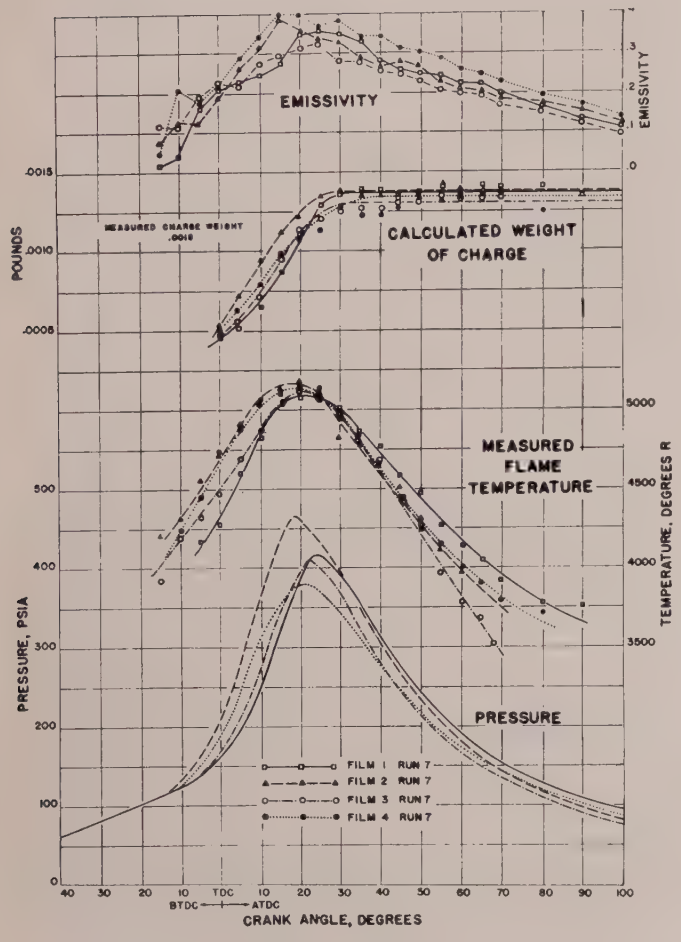


Fig. 7 – Data showing cycle-to-cycle variation

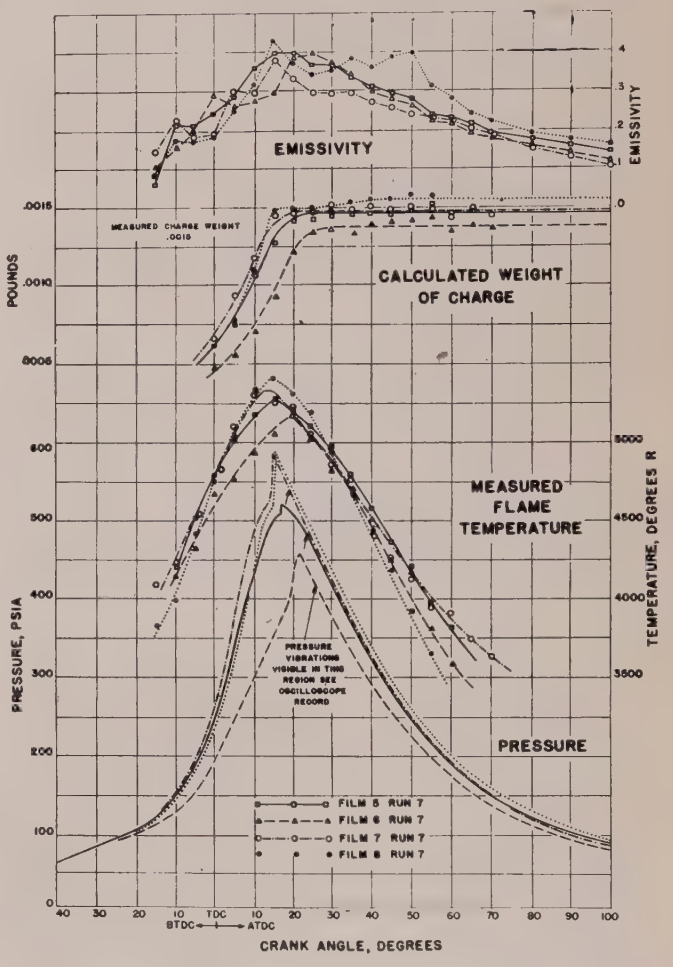


Fig. 8 – Engine data under knocking conditions

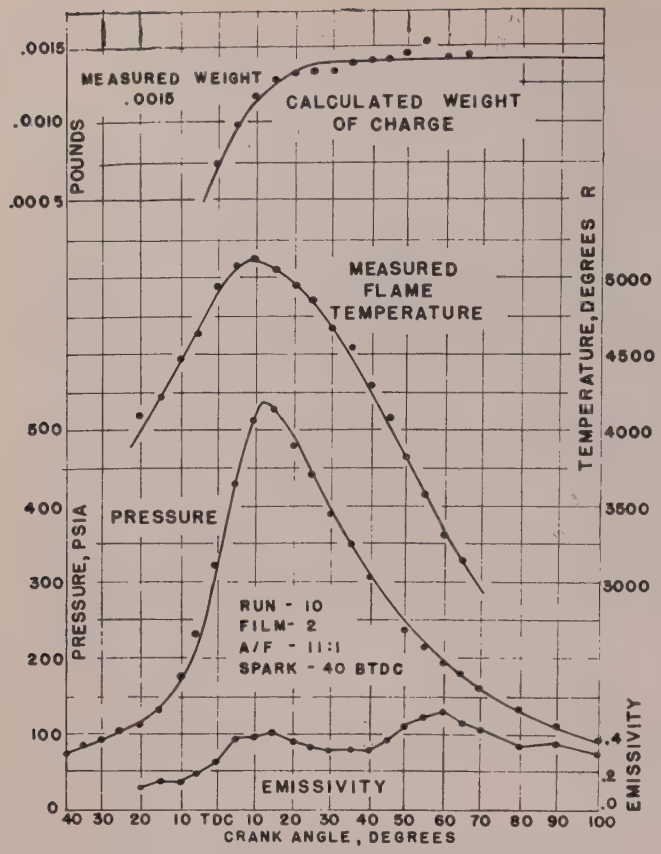


Fig. 9 – Engine data taken with ignition timing of 40 deg btc

but the fact that this constant value is less than the measured weight of the charge inducted is disturbing. This discrepancy could be caused either by the measured temperature's being too high, or by the fact that the engine used had considerable valve overlap, and as a result a portion of the mixture might have passed out through the exhaust valve previous to the compression stroke. Comparison of the weight of charge in the cylinder with the data of Chen and Beck[6] yield values that are approximately the same. There is, however, the difference in engines to consider, inasmuch as one engine was an L-head and the other was an overhead-valve engine. Furthermore, there is the difference in compression ratio to consider: 6/1 for the engine used in these studies and 5.6/1 for the engine used by Beck and Chen. These factors preclude direct comparison, but it is believed that the temperature values shown represent a reasonable value for the average temperature of the burned gases.

As was previously mentioned, considerable cycle-to-cycle variation was noted in the engine. Attempts were made to minimize these variations by proper choice of a spark plug and by insuring thorough mixing and vaporization of the fuel and

[6] See pp. 503-513 of this issue.

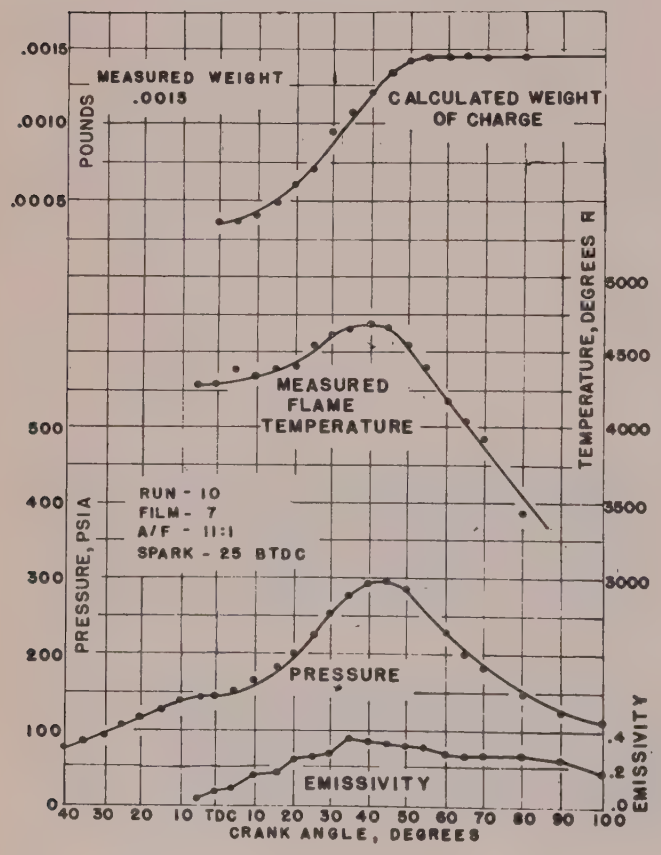


Fig. 10 – Engine data taken with ignition timing of 25 deg btc

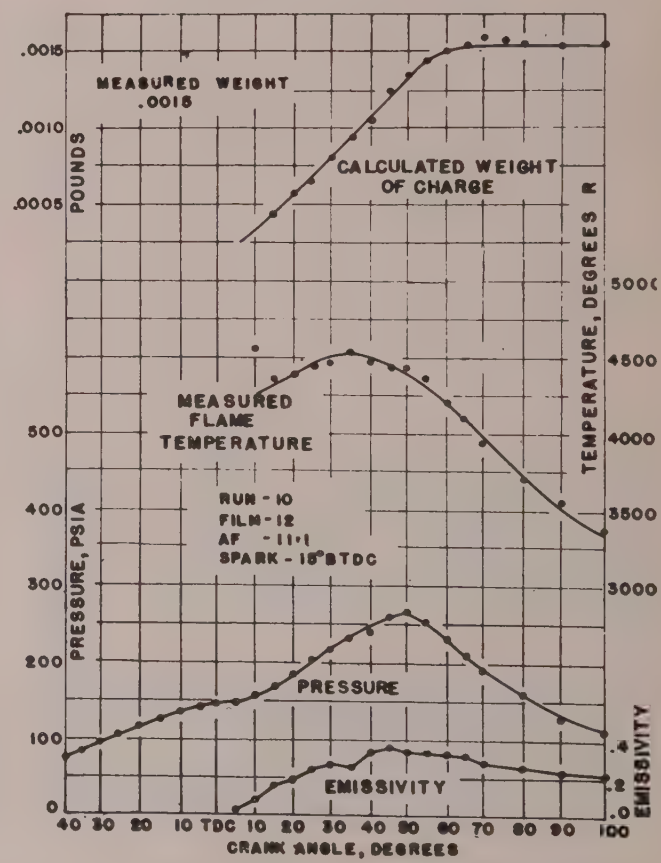


Fig. 11 – Engine data taken with ignition timing of 15 deg btc

air previous to their introduction into the cylinder, but considerable variation remained. The use of a shrouded valve was not attempted.

In an attempt to evaluate the reasons for these cycle-to-cycle variations, two individual cycles were studied. Data for these cycles were taken only a few moments apart under presumably constant operating conditions and are presented in Fig. 17. It can be seen in Fig. 17 that the maximum pressures varied by about 50 psia between the two cycles. One possible reason for cycle-to-cycle variation might be a difference in charge weights. However, as shown in Fig. 17, both calculated charge weights reach nearly the same maximum value indicating no difference in the weight of charge inducted.

Since the spark was experimentally observed to occur at the same crank angle and since no differences in the calculated charge weights were found, it was reasoned that there must have been a difference in the rate of burning between the two cycles. Consequently, calculations were made of mass burned versus crank angle for the two cycles. The computations were made from the pressure-time diagram as suggested by Withrow and Rassweiler[7] and by use of the measured temperatures.

[7] See SAE Transactions, Vol. 33, (May) 1938, pp. 185-204: "Motion Pictures of Engine Flames Correlated with Pressure Cards," by G. M. Rassweiler and L. Withrow.

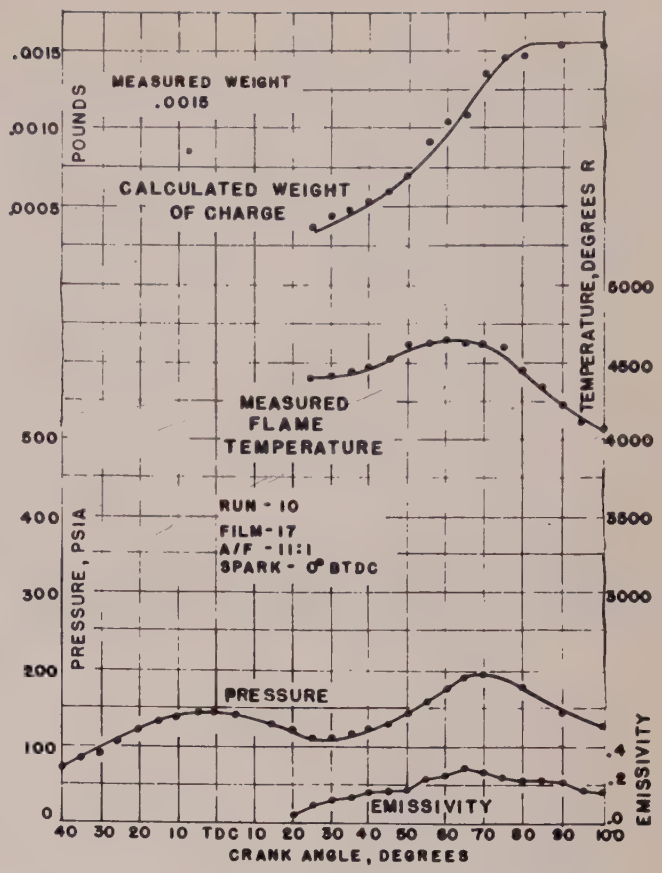


Fig. 12 – Engine data taken with ignition timing of 0 deg btc

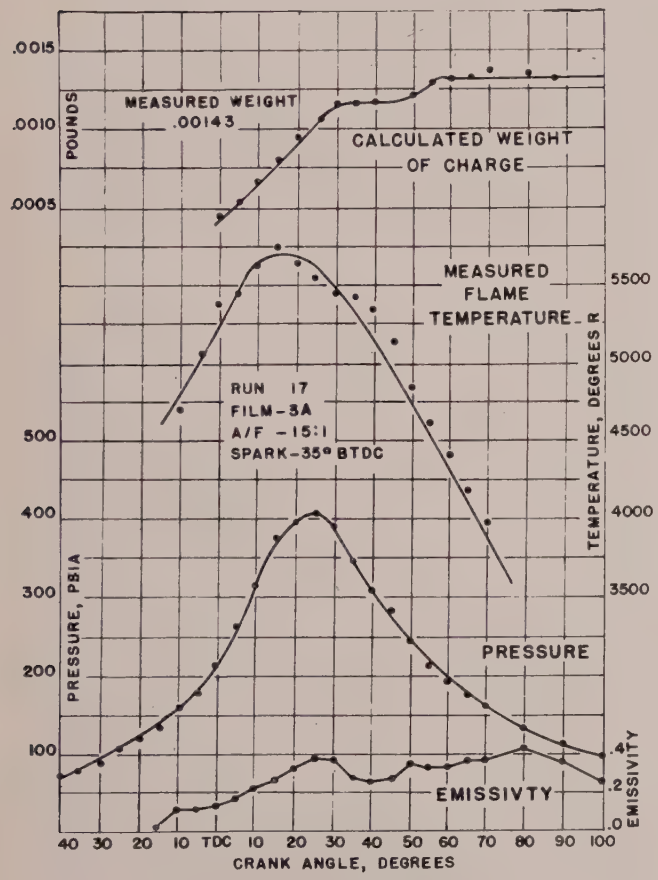


Fig. 13 – Engine data taken with an air/fuel ratio of 15/1

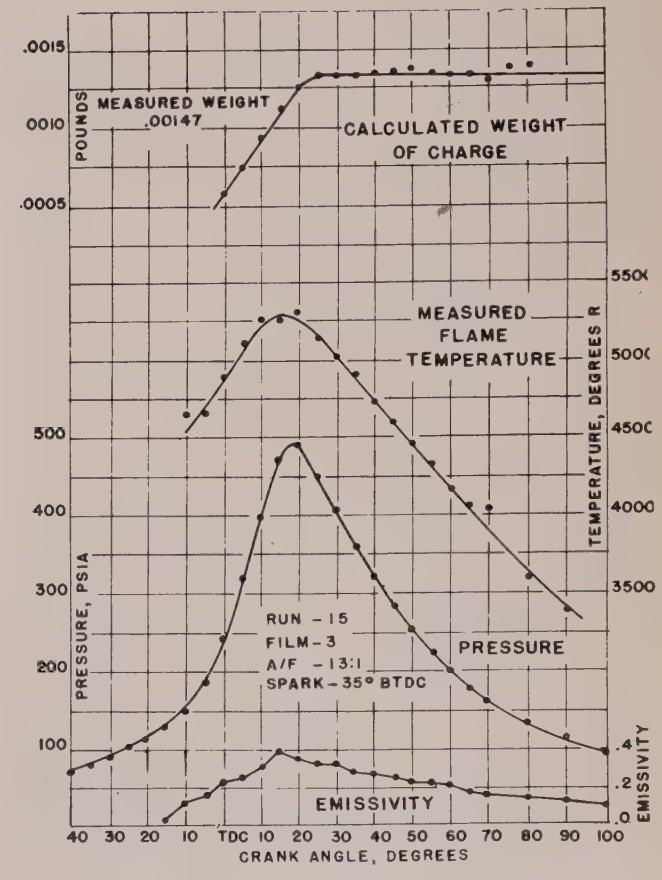


Fig. 14 – Engine data taken with an air/fuel ratio of 13/1

The measured temperatures were used as follows: Using the average value of the calculated charge weight after it had reached an essentially constant value and the measured pressures and volumes, the temperature of the unburned charge prior to spark was calculated. The results of such calculations were plotted against pressure and extrapolated to give the temperature of the end gas. It was next assumed that the measured temperature represented the average values of the temperature of the burned gases only. Justification for this assumption is found in the fact that the calculated charge weights do reach a constant value. Using these assumptions there are four unknowns – the weights and volumes of the burned and unburned gases. Application of $PV = WRT$ to both the burned and the unburned gases and recognition that the sums of the weights and volumes of the burned and unburned gases must equal the total weights and volumes permit one to determine the per cent mass burned as a function of crank angle. The equation used was:

$$W_b = \frac{P\ V_T - W_T R_{UU}}{R_b T_b - R_U T_U}$$

where the subscripts T, U and b indicate that the quantities apply to the total, unburned, and burned portions of the charge. Fig. 18 shows the results of these computations. Reasonably good agreement is shown between the two techniques.

Further results of these computations are shown in Fig. 19. The ordinate is per cent mass burned, and the abscissa is crank-angle position which, neglecting torsional oscillations, is a function of time. The slopes of the curves are related to the rate at which the fuel-air charge is consumed by the advancing flame and proportional to the rate at which heat is released. It will be noted that the rate of heat release is related to the chemical reaction rate of combustion by a factor proportional to the area of the flame front. As can be seen from the curves, the slopes are almost identical, but the lower portion of the curve representing the early part of the combustion process shows marked differences.

Close examination of the curves reveals that the major difference in these two cycles is a result of phenomena occurring early in the combustion process. After 10% of the mass is consumed, little difference exists in the rate at which the fuel-air charge burns. It appears significant that the propagation of the combustion process just after ignition by the spark is the controlling factor in cycle-to-cycle reproducibility and that an improvement

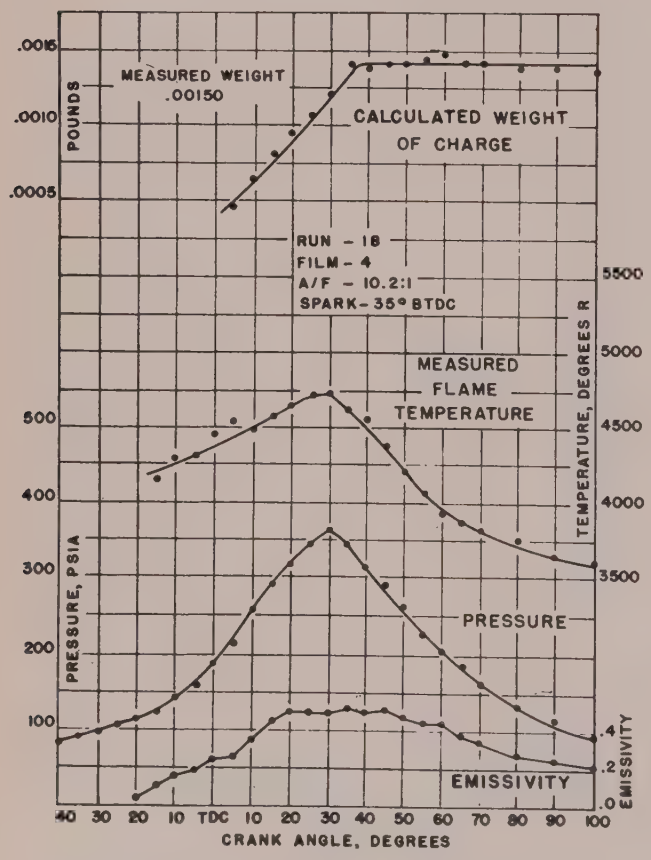


Fig. 15 – Engine data taken with an air/fuel ratio of 10.2/1

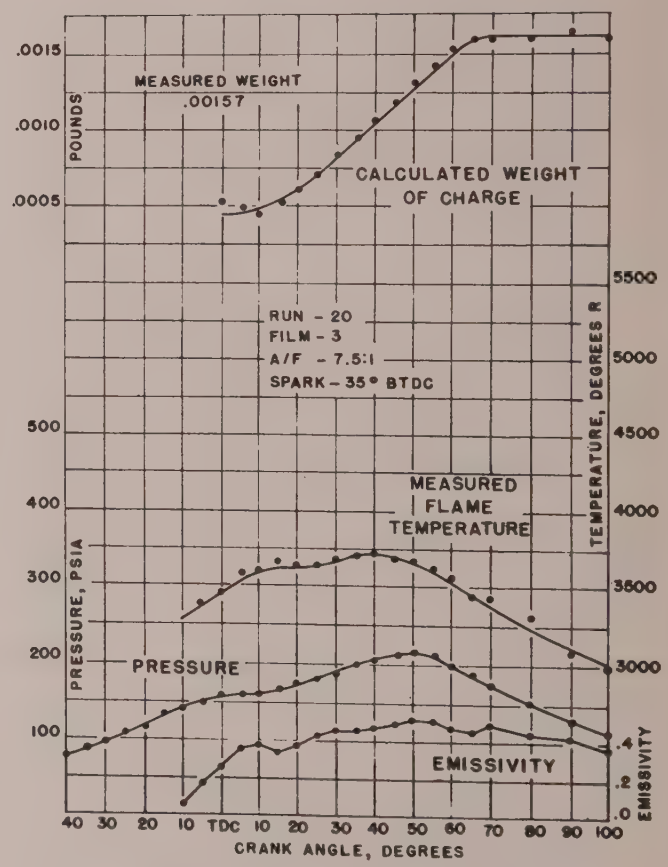


Fig. 16 – Engine data taken with an air/fuel ratio of 7.5/1

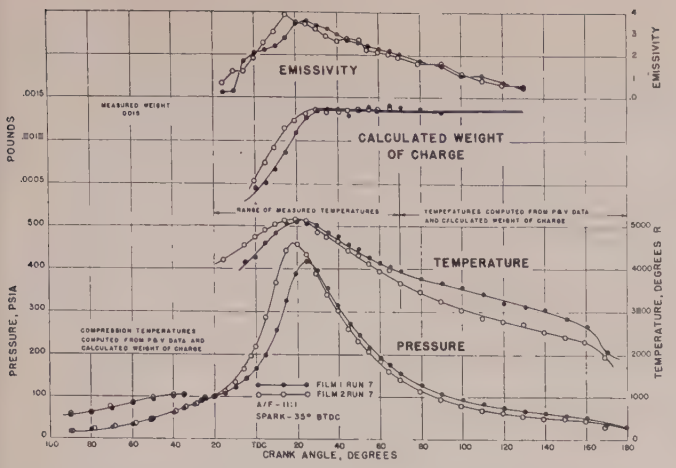


Fig. 17 – Pressure and temperature history of engine showing cycle-to-cycle irreproducibility – two separate engine cycles under similar operating conditions

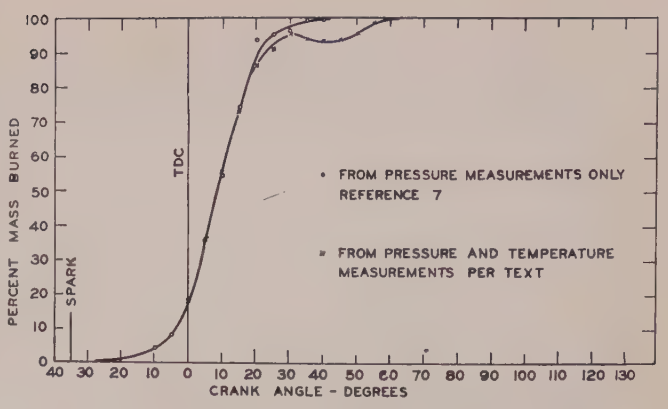

PERCENT MASS BURNED
TDC
SPARK
• FROM PRESSURE MEASUREMENTS ONLY REFERENCE 7
× FROM PRESSURE AND TEMPERATURE MEASUREMENTS PER TEXT
CRANK ANGLE - DEGREES

Fig. 18 – Comparison of techniques for calculating per cent mass burned versus crank angle.

in specific engine performance could be realized by a technique which would result in controlled burning in the time just following ignition by the spark plug.

It is generally considered that the power output of a spark-ignition engine is basically limited by knock. Knock in turn occurs when engine conditions are such that the pressure-temperature-time path of the unburned charge leads to a condition in the end gas where spontaneous ignition results. Knock is more easily obtained under high pressures and temperatures than less severe conditions. As a result, an engine is knock-rated on those cycles which result in the highest temperatures and pressures. In a multicylinder engine, the knocking cylinder is the cylinder on which the engine is rated. If this cylinder has considerable cycle-to-cycle variation, the most severe cycles are the cycles on which the engine is rated. If the cycle-to-cycle variation was reduced so that more of the engine cycles approached the peak pressures and temperatures of the rating cycle, a proportionally greater amount of power could be taken from the engine.

Fig. 19 indicates that to minimize cycle-to-cycle variations some means must be found to control the initiation of burning and the rate at which the chemical reaction proceeds during the first 10% of the combustion period.

Since computation of the mass burned versus crank angle proved to be of interest in studying cycle-to-cycle variations, similar computations were performed for the cycles studied where the variables were air/fuel ratio and spark advance. The results of these computations are summarized in Figs. 20 and 21 where per cent mass burned is plotted as a function of crank angle for different fuel/air ratios and different ignition timing.

These curves show that as ignition timing is advanced, the per cent mass burned advances in a like manner. This is not unexpected, and, in fact, shows that the rate at which the fuel-air charge is consumed is nearly independent of ignition timing. This may be attributed to the slow change in cylinder volume near top-dead-center.

The change in rate of fuel-air conversion with

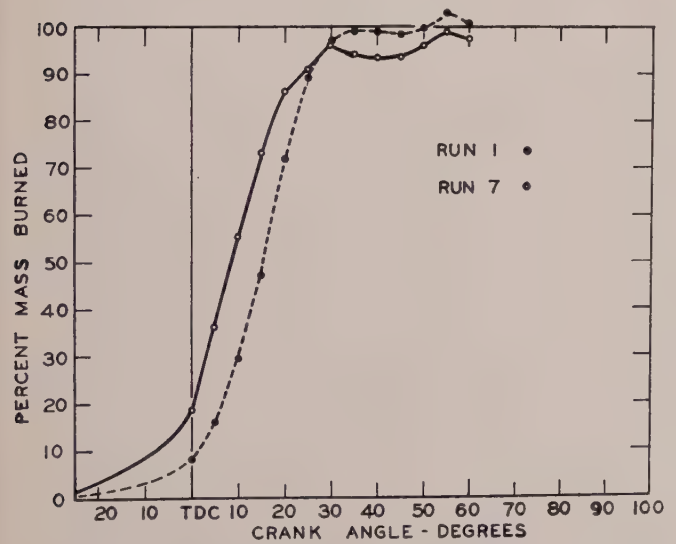


Fig. 19 – Per cent mass burned versus crank angle for two cycles taken under constant operating conditions

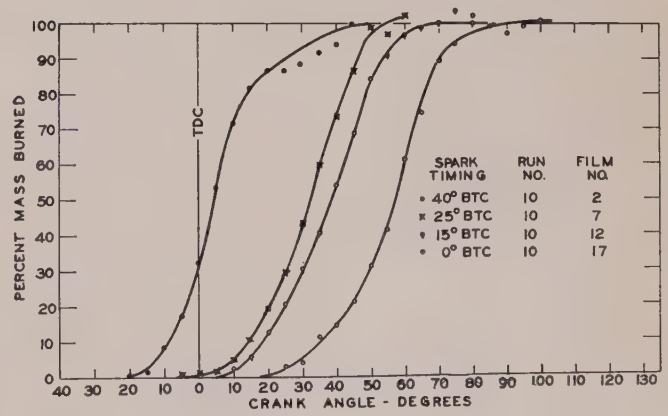


Fig. 20 – Per cent mass burned as a function of spark timing

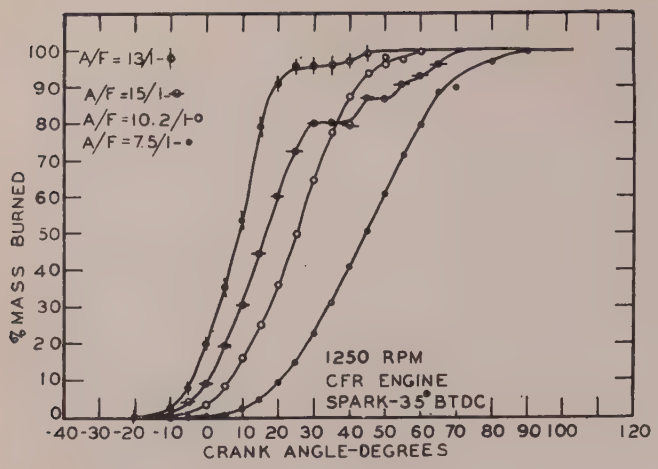


Fig. 21 – Per cent mass burned as a function of air/fuel ratio

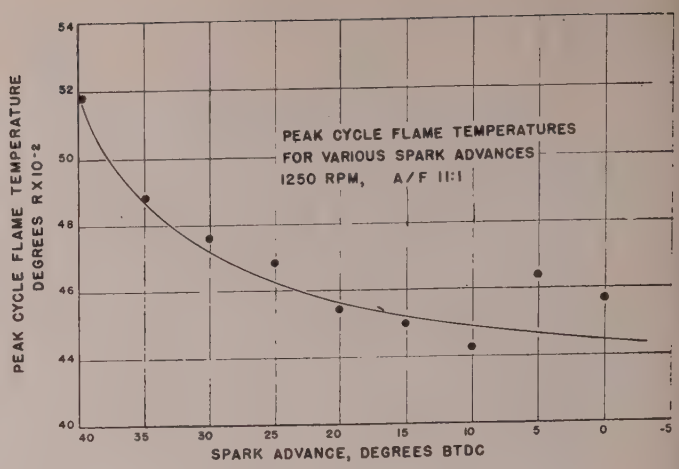

PEAK CYCLE FLAME TEMPERATURE DEGREES R X 10-2
PEAK CYCLE FLAME TEMPERATURES
FOR VARIOUS SPARK ADVANCES
1250 RPM, A/F 11:1
SPARK ADVANCE, DEGREES BTDC

Fig. 22 – Variation of peak temperature with ignition timing

A/F ratio reflects in part a change in chemical conversion rate. The very rich mixtures are not as an effective medium for the propagation of a flame as are the slightly rich mixtures. The greatest reaction rate and highest flame speeds are obtained close to best power mixture conditions.[8]

Since peak flame temperature was considered to be of interest, Fig. 22 presents peak flame temperature in the cycle as a function of spark advance. As would be expected, the highest maximum flame temperature is obtained with the earliest spark advance.

Fig. 23 presents data showing the crank angle at which the peak temperature occurs when the ignition timing is varied. Again, as would be expected, the highest maximum temperatures are observed at earlier crank angles as the ignition timing is advanced.

Fig. 24 presents data showing variation in temperature with air/fuel ratio. Unfortunately, all of the data taken were with rich mixtures. All indications are, however, that the curve would reach a maximum value and then decrease at lean mixtures.

In conclusion, the technique used is simple and seems to give results in reasonable agreement with the equation of state $PV = RT$. The studies presented herein have shown the utility of the technique in studying combustion in an engine with respect to the effect of engine variables on flame temperature and mass burned. It is hoped to investigate the effect of other variables in the future.

Acknowledgment

The authors wish to acknowledge the Wisconsin Alumni Research Foundation, the Ethyl Corp., and the College of Engineering for their generous support of this project.

[8] See NACA TM 857 (1938), "Investigation of Ignition and Combustion Processes of Diesel Engines Operating with Turbulence and Air-Storage Chambers," by H. Petersen.

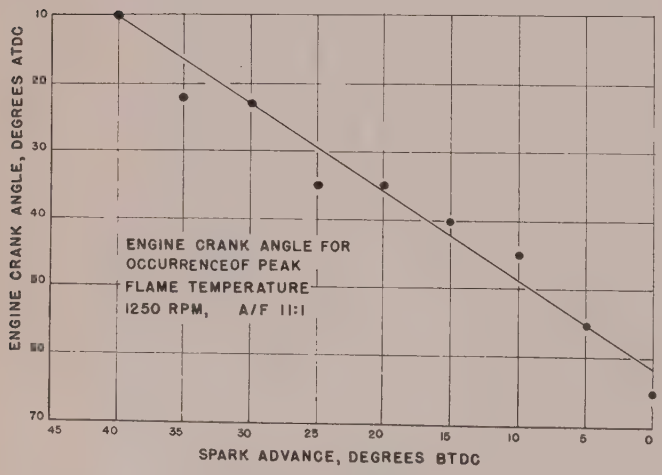


Fig. 23 – Crank angle at which peak temperature occurs when ignition timing is varied

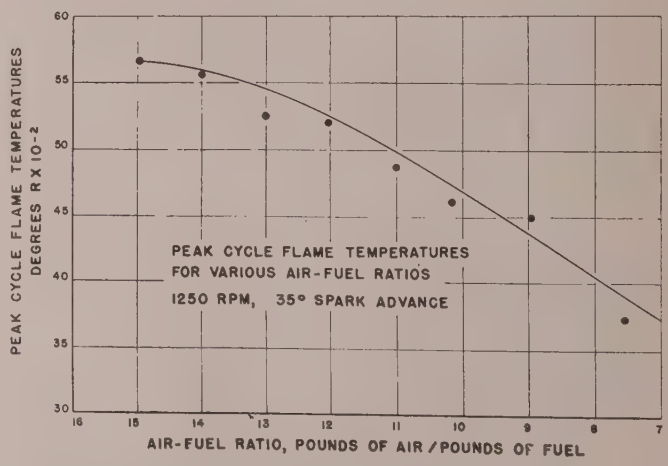

PEAK CYCLE FLAME TEMPERATURES DEGREES R X 10-2
PEAK CYCLE FLAME TEMPERATURES
FOR VARIOUS AIR-FUEL RATIOS
1250 RPM, 35° SPARK ADVANCE
AIR-FUEL RATIO, POUNDS OF AIR / POUNDS OF FUEL

Fig. 24 – Variation of peak temperature with air/fuel ratio

Discussion . . .

. . . of the Chen, Beck, Uyehara, and Myers and the Millar, Uyehara, and Myers Papers

Data from Sound-Velocity and Absorption Test Compared

– P. C. K. Wu and J. C. Livengood
Massachusetts Institute of Technology

FOR a long time engine researchers have hoped for a solution to the problem of measuring the end-gas temperature. Now Chen, Beck, Uyehara, and Myers have demonstrated that the absorption technique can be used effectively, and the further extension of this work is of considerable importance.

However, even at this stage of development it may be useful to compare the results of these absorption experiments with similar data from velocity-of-sound measurements, which were made in the single-cylinder engine shown in Fig. A. It is not within the scope of this discussion to describe in detail the merits of the sound-velocity technique of temperature measurement. For present purposes it is sufficient to state that the special combustion chamber shown was designed with four related objectives:

1. It is desirable from a research standpoint to have a compact end-gas zone. The likelihood of large temperature differences in a small region is not so great as would be the case for an elongated measuring path.

2. It is desirable for reasons of instrumentation that the measuring zone be relatively quiet. Severe turbulence can produce scattering of acoustic energy, and this loss cannot be well afforded.

3. It is necessary, in the case of the two-transducer system shown, to have access to opposite sides of the measuring zone.

4. The multiple gaskets used in the "sandwich" construction provide the needed acoustic barriers, which prevent the sound waves of two megacycles per second from following an alternate path through metal, with the resulting confusion of these signals with the desired gas path transmissions.

The consequence of not providing a small, compact end-gas zone is well illustrated by Fig. 6, where it can be seen that the absorption data ended at 20 deg atc, while the peak pressure was not reached until 35 deg atc. Thus, at the time when the flame entered the measuring path a relatively large portion of the charge was still unburned. An estimate based on the volume of gas ahead of the flame and the pressure and temperature at the instant of flame arrival in the light path shows that at least 15% of the charge was still unburned. This is unfortunate because one of the most interesting periods of the end-gas temperature history is necessarily missing from the record. In view of this fact, would it not be profitable in future development to locate the light path in a smaller and better defined end-gas zone, even if this change should require more modification of the chamber? From the absorption standpoint, a shorter path still can be tolerated when the density is high, which is the case near peak pressure.

A short path has the additional advantage of giving a relatively "local" temperature instead of a spacial "average" value obtained with a long path. However, the foregoing comment does not imply that the "long path" used in the absorption measurements yields a "temperature" that is a true average temperature. On the contrary, the variation of W/W_o as plotted in Figs. 11-14 can only mean that the temperature in the light path was **not** the average temperature of the entire charge, or that the measurement was in error. Incidentally, the terminology in the ratio W/W_o may be misleading. W, which is the calculated ratio pV/RT, where T is the temperature in the light path, should not be called "the weight of charge in the cylinder" because we know that this quantity is constant throughout the valve-closed period, neglecting leakage.

The temperature versus crank angle curves shown in Figs. 11 and 12 represent tests with all conditions identical with the exception of the fuels used. If these curves are superimposed, the curve for 90-octane fuel is above the 50-octane fuel curve throughout the entire period of measurement. Early in the cycle, the difference is 50 deg, and later it rises to about 300 deg. In contrast with this result, data obtained with the M.I.T. sound-velocity method are shown in Fig. B. The fuels used were isooctane, normal heptane, and a 55-octane-number primary blend. The speed was 1200 rpm and the compression ratio was 3.56. There was no detectable detonation, even when using n-heptane. Until a short time before the flame reached the measuring region, the three fuels gave identical end-gas temperatures.

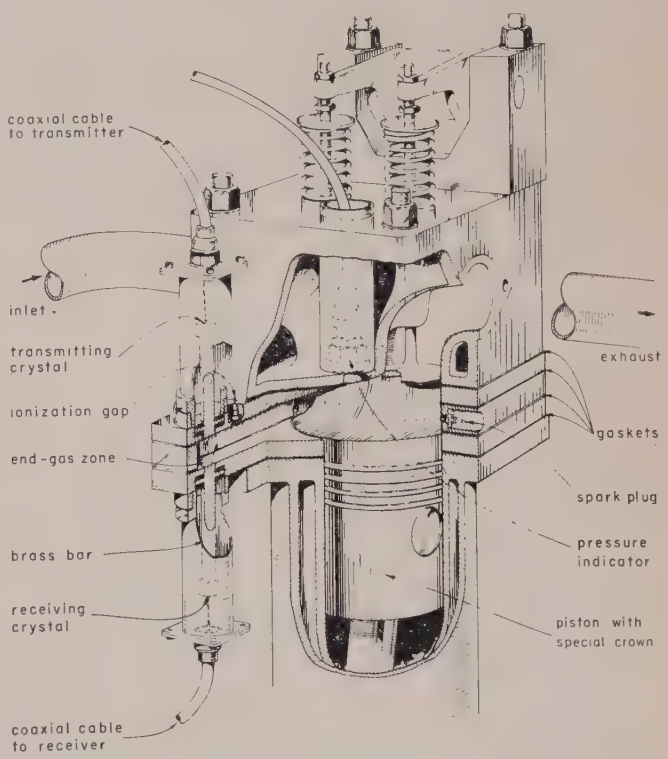


Fig A – Special combustion chamber and cylinder head used in making temperature measurements by sound-velocity technique

However, in the very-high-temperature environment, there is a separation of the curves, with n-heptane giving the highest values and isooctane the lowest. These running conditions were such that the engine was close to incipient knock while running on n-heptane. Consequently, it is likely that precombustion reactions were occurring during the last part of the end-gas history. With this probability in mind, it is not unreasonable to expect that the more reactive fuel-air mixtures should exhibit higher temperatures, since it has been shown – at least semiquantitatively – that the late prereactions are exothermic.[a] This result is in contradiction with the data presented by these authors. We hope that these discrepancies can be resolved after both methods of defining and measuring temperature have been studied further.

In order to end with a more optimistic comment, we present Fig. C. The variable was speed, the fuel was isooctane, and the spark was adjusted to best power at each speed. The trend of higher temperature with higher speed agrees with the presented iodine absorption data. This result is, of course, entirely compatible with established principles of heat transfer.[b] However, those instances where the instantaneous polytropic exponent exceeds the value for reversible isentropic compression require a more explicit justification than is now available.

As a final reminder concerning the magnitude of the measurement problem (regardless of chosen technique), it should be noted that the maximum slope of the 1800-rpm curve of Fig. C represents a time rate of change of more than a half-million Fahrenheit degrees per second!

[a] "Effect of Preflame Oxidation Reactions on Engine Knock," by C. Walcutt, J. M. Mason, and E. B. Rifkin. Presented before the Division of Petroleum Chemistry, 124th Annual Meeting of American Chemical Society, Chicago, Sept. 10, 1953.

[b] See "Proceedings of the General Discussion on Heat Transfer," pp. 397-401; "Heat Transmission in Internal-Combustion Engines," by C. F. Taylor. Pub. by Institution of Mechanical Engineers, London, 1951.

Compares New Data With Earlier Results

– George A. Ball
Ethyl Corp.

COMBUSTION research has been plagued by a lack of reliable temperature data, and in such circumstances any development of a new technique is of major interest. Chen, Beck, Uyehara, and Myers have presented one such development.

Now the first test any new technique must meet is comparison with the whole accumulation of past experience to see if any discrepancies are evident. If we should find any apparent contradictions, then the various data should be examined critically. The authors, in their discussion of Fig. 15, suggest a possible disagreement between their results and the work of previous investigators. It is our belief that a disagreement does exist and that the change in gas temperature with fuel type does not follow the trend

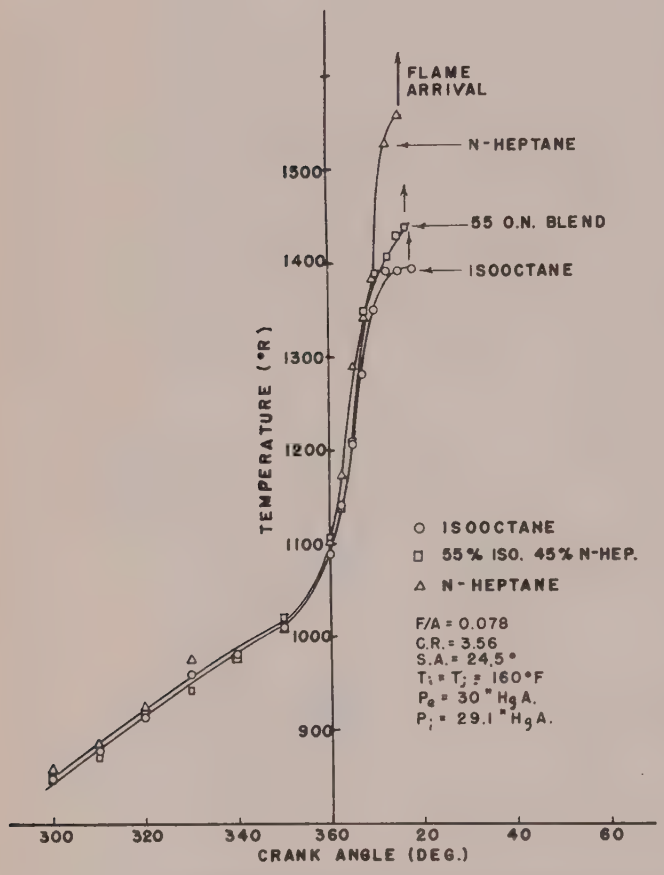


Fig. B – End-gas temperature with different fuels – 1200 rpm, sound-velocity method

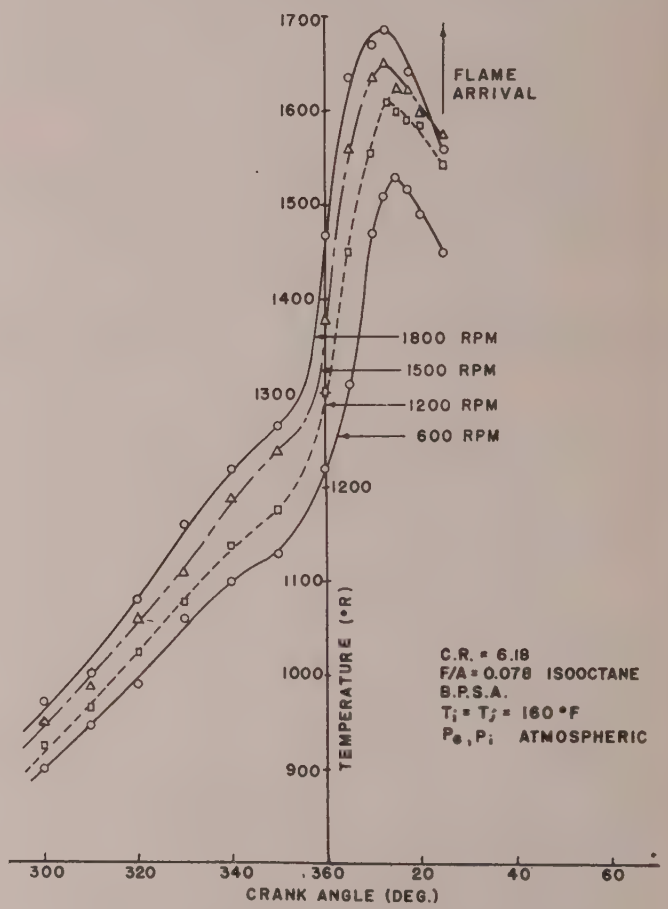


Fig. C – Effect of speed on end-gas temperature – sound-velocity method

of the data reported in Fig. 15, assuming that the temperature is adequately defined by the perfect-gas law. In order to understand our belief, it is necessary to keep two facts in mind. First, published data on primary reference fuel blends show that, under conditions where precombustion reactions are occurring, the greater the proportion of normal heptane the higher the pressure during the precombustion period. Second, according to the perfect-gas law, an increase in pressure, for a particular cylinder volume, is accompanied by a proportional increase in temperature, assuming a constant number of moles. To be sure, the details of precombustion reactions are not yet established and it is not possible to compute the actual change in moles. But all plausible reaction schemes imply a molar change of only a very few per cent at most, which is insufficient to account entirely for the pressure increases observed experimentally. Therefore, we have no alternative but to infer that the precombustion reactions release a net amount of heat and consequently the average temperature in the combustion chamber increases. So, we expect increased pressure and increased temperature with increased normal heptane concentration. Fig. D shows a typical set of pressure data. After 10 deg before top center, we see an increase in pressure level with decrease in octane number, that is, with increase in normal heptane concentration. A corresponding increase in average temperature level is therefore implied. The nearly vertical portions of the curves for the 40% and 50% primary reference fuel blends indicate autoignition of the charge and therefore the termination of precombustion in these cases.

With these two facts in mind let us examine the data from the iodine absorption technique. First the data at 1600 rpm (Figs. 13 and 14) show that both the temperature and the pressure increase as the concentration of normal heptane increases in the region between 15 deg before top center and 20 deg after top center in agreement with our two facts. Of course, these are only two sample runs and they may not be entirely representative of all the data taken. Now when we examine the data at 1100 rpm (Figs. 11 and 12), we find the pressure data to be inconclusive, while the temperature data show a reversal trend. We search, therefore, for a reason to question the data at 1100 rpm. Suppose we look at the plots of W/W_o the relative weight of charge in the cylinder. At 1100 rpm, we consistently find a 20 to 30% increase in this factor up to the point of ignition. Now, if the pressure data are reliable, and the change in the mean molecular weight of the gases is small, this implies a 20 to 30% discrepancy in average temperature from the perfect-gas law. The 1600 rpm data are much better in this respect in that the plots of W/W_o are more nearly constant. With such a large discrepancy in the measured temperature, we do not feel that we can accept the reported effect of fuel type on temperature on the basis of the data reported.

We are unable to follow in detail the discussion on 2-stage precombustion reactions. The usual picture depicts two regions of major heat effects, the heat release of the cool flame and the heat release of the hot flame. Since the hot flame is not under consideration here, there is apparently only one regime in which any measurable heat effect occurs, namely, that of the cool flame. Thus, it is not clear how to account for any reversal in trend based on a competition of heat effects.

Despite our disagreement with some of the results presented in this paper, we do feel, in agreement with the authors, that the method shows promise of being able to solve one of the most difficult experimental problems encountered in engine research.

Temperature Gradients Shown To Occur in Burned Gases

– W. G. Agnew
GMC Research Laboratories

THE authors of these papers indicate that they expect their measured temperatures to correlate reasonably well with temperatures calculated from the perfect-gas law during those portions of the engine cycle before ignition and after the completion of combustion.

On this basis Millar, Uyehara, and Myers have determined the weight of the charge in the cylinder and the per cent mass burned as a function of crank angle. Chen, Beck, Uyehara, and Myers have determined a quantity of W/W_o, which they say is proportional to the weight of charge, and have interpreted their temperature-pressure curves in terms of heat addition and heat loss from the charge.

In the case of the burned-gas temperatures of Millar, Uyehara, and Myers, the agreement between the measured and calculated charge weights is remarkably good under the circumstances. The curves of per cent charge burned determined by two different means also show good agreement – at least in the one case cited.

However, we would not like the reader to have the impression that such agreement is necessarily expected in all cases. It is presumed in most thermodynamic relations that the working medium is homogeneous in its composition and properties. Since, in an engine cylinder, temperature gradients must certainly exist, the temperature computed from the perfect-gas law is a fictitious quantity representing, at best, an average temperature throughout the gas volume. There is no law in nature's book that says this average temperature must agree closely with the temperature measured in any one small portion of the cylinder.

The fact that temperature gradients can be of great importance is illustrated by Fig. E. This figure is taken from a paper by Rassweiler and Withrow.[e] The three curves at the top represent the temperatures of the burned gas – as determined by the sodium-line-reversal technique – at three different locations in an L-head single-cylinder engine. The lower curve is a pressure trace. It can be seen that temperature differences as large as 300 or 400 deg can exist in the burned gases after the completion of combustion. In knocking combustion the difference reached 600 deg.

Further evidence of the existence of temperature gradients appeared in the curves given by Millar, Uyehara, and

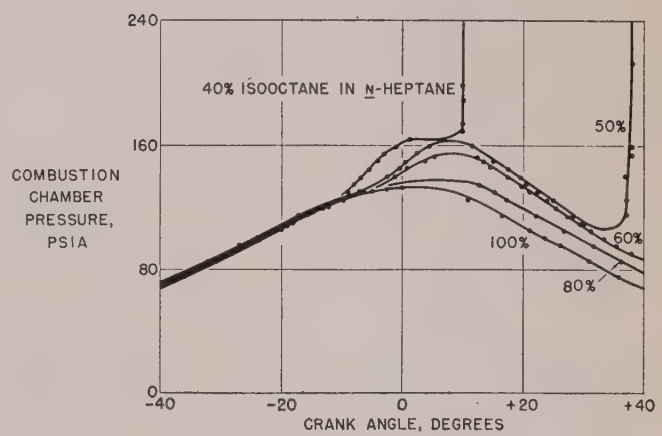


Fig. D – Pressure effect on precombustion – delayed ignition

[e] See SAE Transactions, Vol. 30 (April) 1935, pp. 125-136: "Flame Temperatures Vary with Knock and Combustion-Chamber Position," by G. M. Rassweiler and L. Withrow.

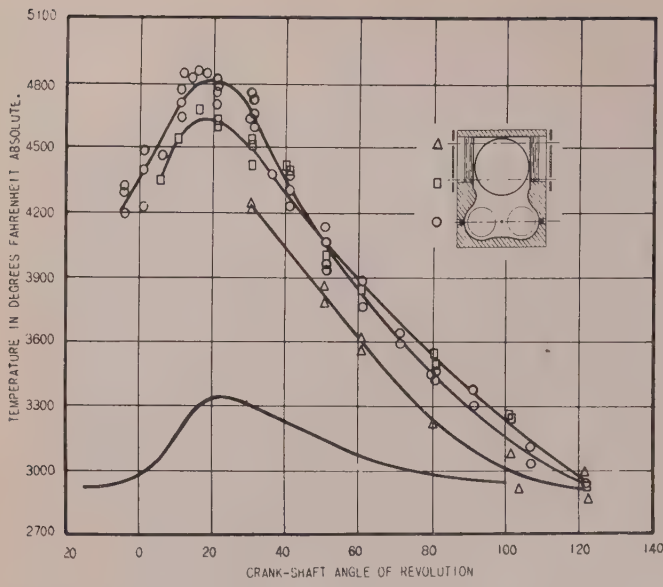


Fig. E – Temperatures measured at three different positions in the combustion chamber under nonknocking conditions

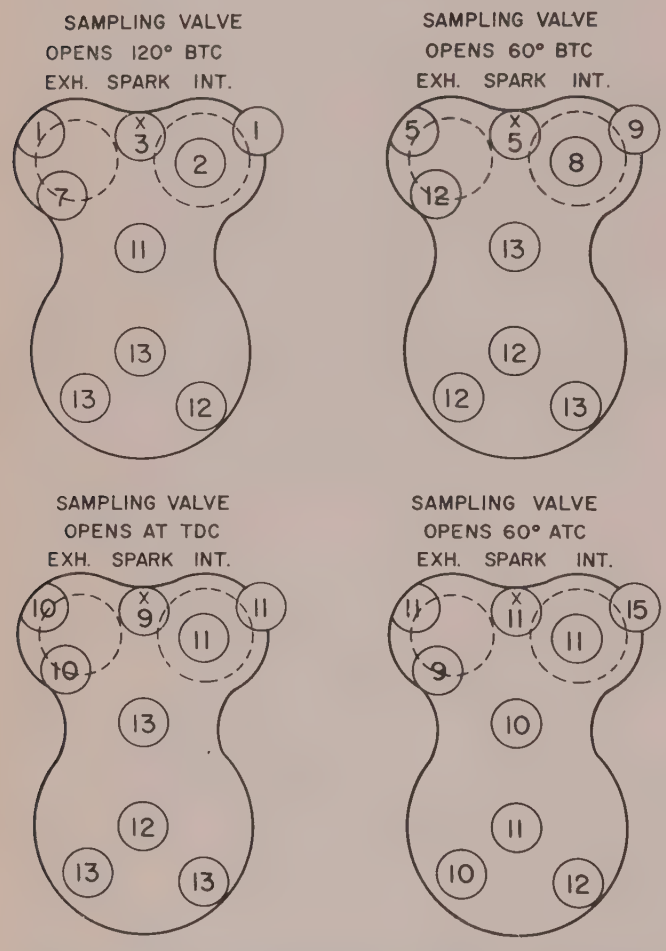


Fig. F – Distribution of residual exhaust gas dilution in single-cylinder L-head engine – full throttle, 1000 rpm, sampling time 60 crank-angle deg

Myers. It was noted in the curves they presented that in almost all cases their measured temperatures reached a maximum 2 to 15 crank-angle deg before peak pressure. This effect was also mentioned in the 1935 paper by Rassweiler and Withrow. The explanation for this observation suggested by them was that the rate of heat loss through the cylinder walls from the burned gases at the firing end exceeded the rate of heat addition by compression during the period in which the local temperature was falling and the average cylinder pressure was rising. Such a heat loss from the burned gases, of course, requires a temperature gradient between the bulk of the gas and the walls of the engine cylinder.

It is unfortunate that engine cyclic variations played such an important role in the data of Millar, Uyehara, and Myers. Under these circumstances it appears that burned-gas temperatures determined continuously offer little advantage over measurements made by a point-to-point method, for which the instrumentation is much simpler.

In connection with the paper by Chen, Beck, Uyehara, and Myers, it should be pointed out that temperature gradients can be an important factor even in the unburned charge. Temperature gradients in this case probably result both from heat-transfer effects and unequal distribution of the residual exhaust gas. A brief study of residual gas distribution in a single-cylinder L-head engine was made in the General Motors Research Laboratories in 1939. Sampling valves were used to draw samples of the unburned charge from a number of different locations in the engine cylinder. The engine was fired in 9 out of every 10 cycles, and samples were taken at various times during the tenth, unfired cycle. The results are shown in Fig. F.

The numbers shown at various locations in the engine-cylinder diagrams represent the per cent exhaust gas dilution at each location. It can be seen that with the sampling valve set to open at 120 deg before top center on the compression stroke a large gradient in the exhaust gas dilution was found. The highest dilution was in the portion of the chamber farthest from the spark plug. At later times in the cycle the distribution became more even.

These data only indicate the type of gradients that might occur. Since the samples were only taken from points near the top of the combustion chamber, they don't give a complete picture.

It is natural to expect that large temperature gradients accompany this uneven distribution of residual gas, since the residual gas is much hotter than the intake mixture.

The gradual equalization of exhaust gas dilution and corresponding temperatures during the compression stroke would help to explain the increase in W/W_o shown in a number of the curves given by Chen, Beck, Uyehara, and Myers. The fact that the rise in W/W_o was more prominent at 1100 rpm than at 1600 rpm indicates that this effect may be related to the spread in end-gas temperatures with fuel octane-number – which also appeared only at the lower speed.

Analysis of these data in terms of exhaust gas dilution would be made considerably easier if the authors had indicated clearly when the engine was knocking and when not.

With regard to the experimental technique of end-gas temperature measurement, we would like to question the authors on one point. It was indicated in the paper that the results were extremely sensitive to the measurement of zero light absorption. In spite of this fact, a measurement of zero absorption through the exhaust gas during the exhaust stroke was considered preferable to the measurement obtained with the light chopper. It would appear that the exhaust gas reference point might be considerably more subject to variation with engine operating conditions. In addition, this technique apparently prevents measurements from being taken in a motored engine – which we feel would provide a better test of the validity of the temperature measurements than any data yet presented.

Ignition Source Probable Cause of Cyclic Variations

– John T. Agnew
Purdue University

REGARDING the comparison of the calculated weight of charge (using the measured flame temperature) and the measured weight of charge, Millar, Uyehara, and Myers have commented that the calculated weight is less than the measured weight. Actually, this is not always the case, even for the few curves they were able to reproduce in this paper. In fact, there appear to be two definite trends evident in the variable spark timing and variable fuel/air ratio series. As is indicated in Figs. 9 to 12, inclusive, when the spark advance is decreased from 40 deg btdc to 0 deg, the ratio of the calculated charge weight to the measured charge weight progresses according to Table A.

This same trend is noticed in the variable fuel/air ratio series as shown in Table B.

Since, of the three experimental variable used to calculate the weight of charge (P, V, and T), temperature is probably the least accurate, this would indicate that the temperatures are too low at low fuel/air ratios and low values of spark advance, and vice versa.

The emissivity of the flame tends to decrease with decreasing spark advance; but, on the other hand, it tends to increase with decreasing air/fuel ratio, as would be expected. Therefore, from the data given, there appears to be no readily apparent explanation for the above trends.

The fact that the calculated charge weight to measured charge weight ratio varies 10% over the range of experimental variables tested should be considered in an estimation of the accuracy of determination of this difficult parameter, temperature.

Regarding the phenomenon of cyclic variations, the evidence is becoming more and more complete that this is due to variation in ignition of the flame. In the recent Fourth Symposium on Combustion, the Bureau of Mines group presented data on methane combustion showing that the shape of the initial flame envelope depended strongly on the gap width and spark energy. Since the shape of the envelope would affect the mass conversion rate, the variation in gap width and spark energy would ultimately appear in the pressure and temperature history of the entire explosion.

These facts plus the knowledge that the spark produced in most engine cylinders is far from being reproducible from cycle to cycle would point to the ignition source as one of the primary causes for the cyclic variations.

Table A – Ratio of Calculated to Measured Charge Weights as Related to Spark Advance

Spark Advance, deg btdc	Calculated Weight/Measured Weight
40	0.0014 /0.0015 = 0.933
25	0.00145/0.0015 = 0.967
15	0.0015 /0.0015 = 1.000
0	0.00155/0.0015 = 1.033

Table B – Ratio of Calculated to Measured Charge Weights as Related to Fuel/Air Ratio

Fuel/Air Ratio	Calculated Weight/Measured Weight
15/1	0.00133/0.00143 = 0.930
13/1	0.00133/0.00147 = 0.905
10.2/1	0.00142/0.00150 = 0.947
7.5/1	0.00162/0.00157 = 1.032

Need Uniform, Well-Mixed, Well-Vaporized Charge

– Max M. Roensch
Ethyl Corp. Research Laboratories

WITH this technique of simultaneous measurement of the temperatures and pressures within the cylinder during combustion, Millar, Uyehara, and Myers have furnished the engine designer with factual information of value. For example, the data indicate that during knock the mechanical loads due to high pressures are a more important factor than the thermal loads due to increased temperature. The peak pressures are increased from 450 to 600 psi, or an increase of one-third, while the temperatures are increased from 5150 to 5250, or only 2%.

The authors have commented on the problem of cyclic variation in the pressure rise after ignition, which they have encountered in their test work. We have experienced the same difficulties in a similar engine, and the results of our tests may be of value to others in this field. The tests at the University of Wisconsin were run without shroud on the inlet valve. We have found that the use of a shrouded intake valve was vital in that the direction of the incoming charge, as controlled by the shroud, must be such as to create a swirl in the combustion chamber if cyclic variation was to be minimized. Even with the best location of the shroud, cyclic variation was not satisfactory until the fuel was properly vaporized. Using an intake air temperature of 140 F it was found that isooctane, which boils at 211 F, was not as good as cyclopentane, boiling at 121 F. When the intake air temperature was raised to 380 F, excellent results were obtained with a variation in peak pressure of only 50 psi. These tests indicate that not only must the charge be uniform in its air/fuel ratio as inducted into the engine, but turbulence within the combustion chamber must also be sufficient to ensure a homogeneous charge if uniform performance is to be obtained from cycle to cycle.

Chen, Beck, Uyehara, Myers Closure to Discussion

THE authors offer their congratulations to Messrs. Wu and Livengood for also finding the solution to a difficult experimental problem. The technique and data presented are naturally of considerable interest since they provide an opportunity for direct comparison of results obtained by two different techniques. It is hoped that close cooperation between the groups using the two different techniques will help to determine just what does happen to the end gas in an operating engine. The simultaneous use of both the iodine absorption technique and the velocity-of-sound technique on an engine would certainly be an interesting experiment.

The authors well realize that their combustion-chamber configuration did not permit viewing of the very last part of the charge to burn. The location of the light path is inevitably a compromise between length of light path, minimum alteration of combustion-chamber configuration, and portion of end gas viewed. For these initial tests the authors made a very deliberate choice of a fairly long light path. It is planned to alter the location of the light path in future studies.

With regard to the data presented by Messrs. Wu and Livengood the authors are glad for confirmation of the expected trend of higher temperatures with higher engine rpm. With regard to the data presented for different octane-

number fuels, it should be noted that the information presented in Fig. 15 of the text indicates that higher temperatures are obtained with higher octane fuels when the comparison is made at the same pressure. The authors feel that Messrs. Wu and Livengood have been somewhat premature in stating that their data disagree when no valid comparison has been made. Unfortunately, no pressure data were presented by Messrs. Wu and Livengood and the authors are, therefore, unable to make a valid comparison for their data.

Mr. Ball certainly is correct that any new technique must be tested by comparison with our accumulation of past experiences. It is also axiomatic, however, that if the comparisons are to be valid they must be made under identical conditions. For noncatalytic or self-catalytic chemical reactions identical conditions will only exist when the two items being compared have identical pressure-temperature-time histories. It would seem necessary, therefore, to establish the equality of the temperature-pressure histories before making any comparisons.

Mr. Ball presents no information with his Fig. D concerning the time relationships. The data do appear to have been taken on a motored engine. Since other data on motored engines reported by coworkers of Mr. Ball have usually been taken at less than 1000 rpm it seems safe to assume that the data were taken at no higher rpm than the authors' data.

In a motored engine the gases are compressed by the piston alone. In a fired engine the gases are compressed by both the piston and the burned gases. It follows, therefore, that the gases in a motored engine run at the same or lower rpm as a fired engine cannot have the same time history as the end gas in a fired engine. This is a very important point, for it also follows that the pressure-temperature histories are not the same in a fired and motored engine.

Confirmation of these thoughts will be found in the fact that the end gas in a fired engine is compressed through an effective compression ratio appreciably greater than the autoignition limited compression ratio for the same fuel at the same engine rpm. The authors would reach the conclusion that it is impossible to obtain similar pressure-temperature-time histories in a fired and in a motored engine operated at the same rpm and that any comparisons made are, therefore, questionable. They would, futhermore, offer the comment that more nearly comparable temperature-pressure-time histories (and therefore more nearly comparable data) would be obtained by operating the motored engine at an appreciably higher rpm than the fired engine. To illustrate the point further, the rate-of-pressure-rise in a fired engine is several times that of a motored engine at the same rpm. Certainly knock (and probably precombustion reactions) is markedly different in an engine running at 1000 rpm with a rate of pressure rise that is low compared to an engine running at 3000 rpm, for example.

The authors might also point out that they made comparisons similar to those suggested by Mr. Ball and because of the lack of similarity of results they met together one year after taking the original data to recheck the results. The rerun data confirm the original data and the authors believe that it is the lack of similarity of the pressure-temperature-time histories between published motored engine data and their data that causes the seeming discrepancy rather than the technique. It should also be noted that the different conclusions are being drawn from the measurement of two different properties, that is, pressure and temperature, and that either a change in number of moles (usually considered small) or pressure or temperature inhomogeneities might also cause the seeming discrepancy. Perhaps future data will help to resolve this seeming discrepancy.

Regarding Mr. Agnew's question on the determination of zero absorption, the exhaust portion of the cycle was used to minimize the possibility of error. It was determined experimentally that there was no absorption by iodine during the exhaust stroke. This zero reading would then be correct if no changes occurred in the electro-optical system during a single cycle. Use of the light chopper as zero reference requires that the electronic and optical system remain constant for a period of time longer than one engine cycle. If the light chopper reading were automatically recorded immediately before and after each cycle, both methods of zero absorption determination would have about the same reliability. Measurements of compression temperatures can be made on a motored engine, although the results may be somewhat less reliable than those of a fired engine.

Millar, Uyehara, Myers
Closure to Discussion

NUMEROUS attempts were made during the course of this research program to reduce cycle-to-cycle variations in combustion-chamber pressures and temperatures. Many combinations of air and mixture temperatures were tried but with not too much effect. It was also found that, when the mixture temperature became too high, sodium salts were precipitated in undesirable amounts on the intake manifold.

Various types of spark plugs were also tried in an effort to reduce cycle-to-cycle variations. Although some effect was found the variations were still large. Unfortunately, a shrouded valve was not tried.

The authors agree, with some reservations, to Mr. Roensch's conclusion that the mechanical shock of knock is of more importance than the thermal shock. It must be remembered, though, that the thermal shock is a function of both temperature and the heat-transfer coefficient. The present data do not provide information on the values of the heat-transfer coefficient and, consequently, the comparison between mechanical and thermal shock made by Mr. Roensch may not be entirely valid.

With regard to the temperature gradients mentioned by Mr. Agnew the authors well realize the existence of these gradients and agree that the measured averaged temperature need not (and in fact with most techniques does not) agree with the temperature computed from the perfect gas law. In this connection Mr. Agnew has not presented a complete story on his Fig. E, for it is probable that the temperature gradients in the line of sight of the measuring instrument were just as great as the temperature gradients measured at different locations. Thus, the temperature values plotted in Fig. E are themselves average values and therefore subject to the same comments made by Mr. Agnew concerning the authors' data. The situation can be summarized by the statement that, in systems not in thermal equilibrium, measured values of temperature have no significance until the technique and spatial temperature distribution is specified. The authors do not feel it of importance both from the standpoint of their data and the data of Fig. E (the two techniques were very similar) that there was unexpectedly good agreement between the measured and calculated temperatures.

The authors disagree with Mr. Agnew as to the relative value of point-by-point and instantaneous measurements. If one is interested in the overall performance of the engine, point-by-point measurements are certainly the easiest to make and will present an overall picture of the performance of the engine. On the other hand, if one is interested in the vagaries of the combustion process, it is clear that instantaneous measurements, although appreciably more difficult to make, are more useful. For example, the information presented in Fig. 19 could not be obtained from point-by-point measurements.

In conclusion, the authors have no explanation of the trend shown in the ratio of charge weights. The sodium-line reversal technique has come under criticism as indicating too high temperatures. An indicated trend could result from this or from the previously mentioned problems of spatial averaging of temperatures.

The Chrysler PowerFlite Transmission

W. R. Rodger and A. J. Syrovy, Chrysler Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1954.

FROM the earliest days of the automobile, engineers have dreamed and worked to produce a device that would overcome the shortcomings of the internal-combustion engine in its torque-speed relationship. For 40 years their success was sporadic, but, in recent years, they have succeeded in developing automatic transmissions that are practical and dependable. Within a few years, the self-shifting transmissions have become so firmly established in the passenger-car field that, today, conventional, hand-shifted transmissions have become a rarity in many lines of cars.

However, contrary to developments in other components of automotive vehicles, there does not yet seem to be a basic design trend, which all car makers will eventually follow. There are boosters for the step-ratio type of transmission and there are eloquent proponents of the torque converter of the infinitely variable type. The controversy between the proponents of 2-, 3-, and 4-speed boxes shows few signs of being resolved. One would think that, by this late date, some pattern would have developed, wherein the lower-priced, light cars would tend to one type in the interests of economy, with the luxury cars taking a course toward added performance. However, it seems that such is not the case. The future will be as interesting, then, as the past.

In this evolution under the floor boards, our company has been very active. One of the early steps in that direction was Chrysler's pioneering in 1934 of the automatic overdrive. This was far from an automatic transmission, but it was a start. In 1938 came our "fluid drive." This hydraulic coupling eventually made practical the first modern automatic transmissions. Finally, torque con-

THE design and construction of the PowerFlite automatic transmission are described by the authors.

It is of the torque converter type, some models being water-cooled, while others are direct air-cooled.

Details of the hydraulic controls are explained, including the one-piece shift valve and the shuttle valve for controlling closed-throttle shifts.

It is claimed that this transmission has relative simplicity, light weight, and smoothness of operation.

The Authors

W. R. RODGER (M '43) is with Chrysler's mechanical laboratories where he has held a position since his graduation from Chrysler Institute of Engineering. He also has an engineering degree from Clarkson College of Technology. During World War II, Mr. Rodger worked on experimental vehicles for the Army Ordnance Department.

A. J. SYROVY (M '44) has been with Chrysler engineering since 1931. He received his mechanical engineering training in Czechoslovakia. After coming to this country, he worked with International Harvester and Dodge Brothers in Detroit. During the war, Mr. Syrovy performed design work for the Army Signal Corps.

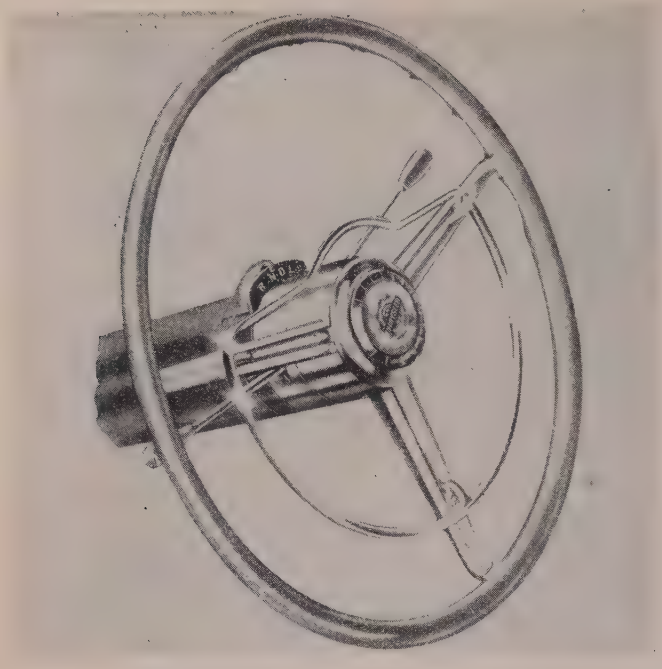


Fig. 1 – PowerFlite range indicator

version was added to "fluid drive" with the introduction of "fluid torque" and "hy-drive." Because these latter types allow no-shift driving while maintaining a degree of manual control, they come under the heading of semi-automatic transmissions.

Smoothness has been the byword throughout the history of hydraulic drives at Chrysler. Therefore, when it was decided to produce a fully automatic transmission, the emphasis was logically placed on this very desirable characteristic. The second requirement was relative simplicity, if the word "simple" is permissible in this connection at all. The performance of the Chrysler PowerFlite transmission is an eloquent reminder that the goal of smoothness has been achieved. Its light weight and relatively few parts attest to the fact that it is among the least complex in the field.

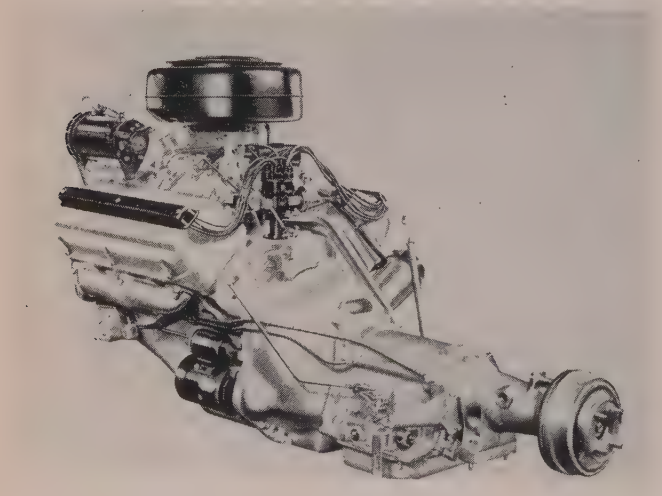


Fig. 2 – PowerFlite installation

The Transmission

The PowerFlite transmission consists of a 4-element torque converter supplemented by two planetary gearsets hydraulically controlled by a multiple-disc clutch and two bands. These provide, along with neutral, a driving range and a low and reverse. These functions are easily selected by a steering column lever, which has a simpler and more convenient shift pattern than any other now available. Fig. 1 shows what a driver sees when he enters a PowerFlite-equipped car. The selector lever gating arrangement, which will be discussed in detail later, is such that all selections can be made by feel without looking at the indicator. This permits the driver to rock his car while he pays the strictest attention to his driving. The complication of a parking position found on most other automatic transmissions has been eliminated, and a very powerful internal-expanding parking brake is mounted on the rear of the gearbox. This brake has been used on our other vehicles as well, and has proved to be very dependable, while retaining emergency braking completely independent of the service brakes.

Fig. 2 shows the relatively neat and compact powerplant assembly that results from the combination of the Chrysler FirePower V-8 engine and the PowerFlite transmission. This transmission is used on the Dodge, De Soto, and Chrysler, on both 6-cyl and V-8 models. The cars vary in weight from 3600 to 5400 lb, and in engine displacement from 230 to 331 cu in. Axle ratios range from 3.36/1 up to 3.9/1. Two different converter sizes are provided: 11¾ and 12½ in. In some models, the transmission and converter are water-cooled; in the others, they are aircooled.

Range Selection

A very confusing situation not unlike that of the 1920's exists today. How well one remembers when every car maker had his own idea on the proper shift pattern! Today's situation is very similar. Refer, if you will, to Fig. 3 and see what automobile drivers have to contend with.

Here are the various indicators together with the standard 3-speed "H." The first automatic transmission had the pattern in group B. This requires shifting through low to reach reverse. Today, the control for this transmission is gated in four different ways. Four other transmissions that followed used this same general arrangement with the parking position added as shown in group C. There, the control is gated in three ways.

Along came a sixth transmission (D) with yet another arrangement wherein reverse was put adjacent to neutral, a much safer place. The parking position was retained.

The PowerFlite transmission has a still different

arrangement, although very similar to the last one mentioned. Its pattern eliminates the parking position and forms an "H" pattern like the conventional standard "H" with second removed.

Here are the advantages of the PowerFlite arrangement:

1. The neutral position can be easily detected, even in complete darkness, simply by moving the lever against the upper stop. This is doubly important because neutral is the only position in which the engine may be started.

2. The drive position is easily reached simply by moving against the lower stop. Hence, the most used positions require no lever manipulation other than movement between the two stops.

3. The shift from neutral to reverse does not require going through a forward gear, with the danger of the car leaping forward when idling fast.

4. All selections can be made by feel with maximum attention being paid to the road. The convenience and safety of this while rocking the car have been mentioned.

5. The parking position is eliminated. The hand brake behind the transmission provides maximum holding power and will, in fact, hold in any range with the engine at wide-open throttle.

It is our opinion that it would be desirable to standardize on the shift pattern for automatic transmissions. This pattern should be as simple as possible and still meet the requirements of all types, and also be as nearly similar as possible to the conventional manual pattern.

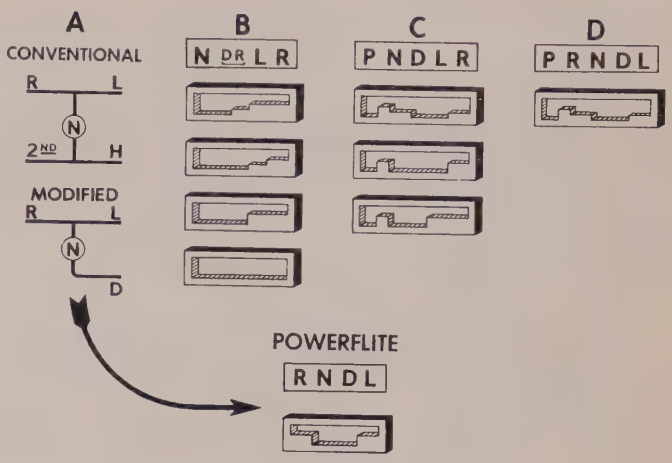


Fig. 3 – Competitive indicator and gating patterns

The Torque Converter

As previously indicated, the torque converter is a single-stage, 4-element unit comprising an impeller, a turbine, and two stators. Fig. 4 shows a cutaway of the water-cooled unit. We have two converters of different diameters: 12½ in. for all De Soto and Chrysler 6- and 8-cyl models, and 11¾ in. for all Dodge models. The smaller converter is also used on Plymouth Hy-Drive. Since the converters differ only in size, this discussion will be mostly about the larger unit.

The primary goal in the development of these converters was high torque multiplication with low engine stall and overrun speeds. From the point of view of manufacturing, these converters had to be suitable for a very high production rate, have minimum weight, and the smallest possibility of external leaks. It was also necessary to get a high degree of standardization of internal parts for all our various requirements.

In these converters, the impeller is attached rigidly to the crankshaft, eliminating the flexible drive plate commonly found in this construction. Early in the torque converter program, it was found that a flexible drive caused excessive engine main bearing rap. The explanation for this lies in the added inertia of the system, which resists crankshaft deflection. The impeller blades are straight and radial and are assembled directly into the converter shell by brazing. This eliminates intermediate stampings and provides reinforcement for speeds up to 7000 rpm.

The turbine is also assembled from stampings, with brazed vanes. Both stators are aluminum castings mounted on the reaction shaft. A roller-type internal cam freewheeling unit permits only forward stator rotation. The rollers of these units are individually energized with accordion-shaped wavy springs of stainless steel. The turbine thrust is taken by flanged powdered-metal bushings, and stator thrust by steel-back bronze washers.

Fluid coupling practice is retained in that the

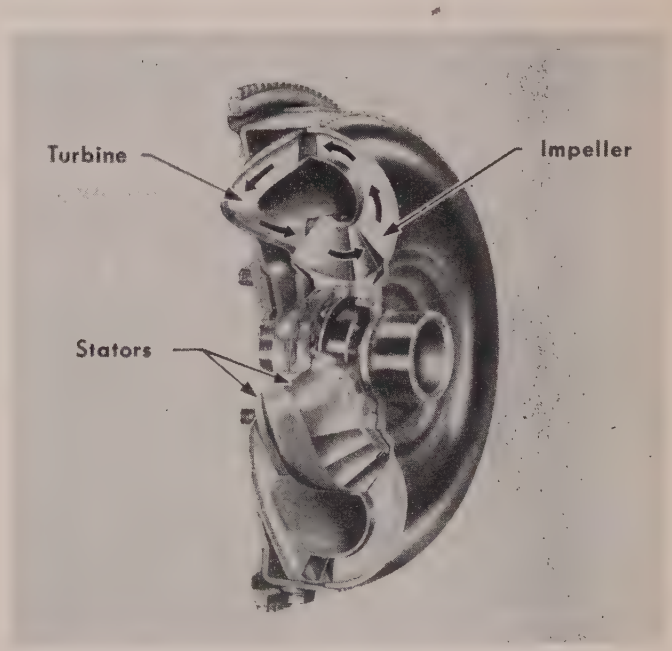


Fig. 4 – Torque converter

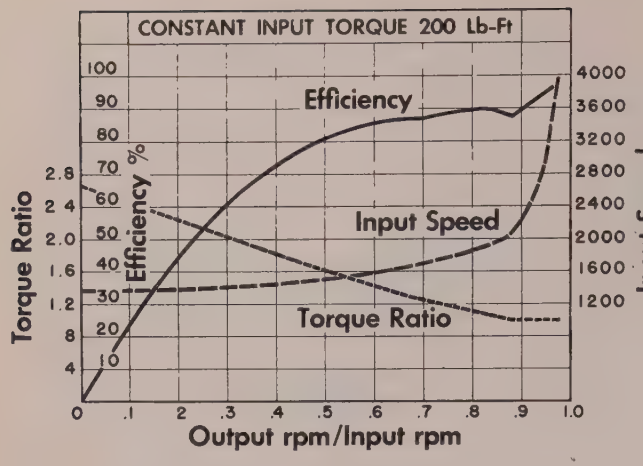


Fig. 5 – Performance of 12½-in. torque converter

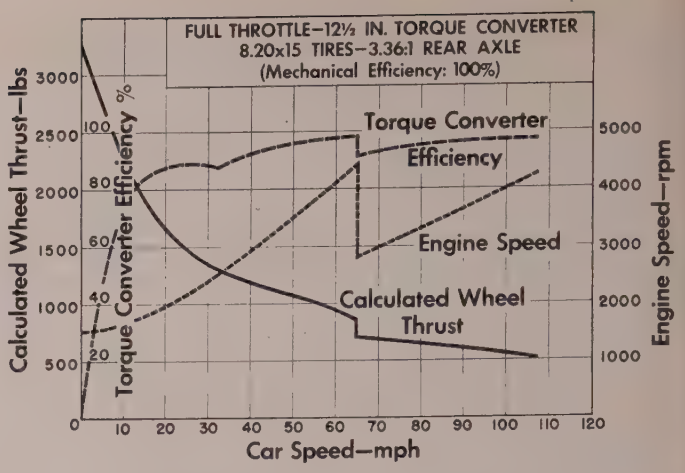


Fig. 6 – Performance of Chrysler V-8 with PowerFlite

assembly is welded around the periphery. Thus, heavy flanges and bolts are eliminated, with resulting lower inertia. Furthermore, a leakproof assembly results. Experience has shown that service requirements are met very satisfactorily by this type of construction.

A torque converter has an unbalanced forward thrust due to the opening through which the turbine and reaction shafts project. During the development period, we had an idea that this thrust could be balanced by relieving the pressure from behind the turbine through venting holes. As occurs so often to the best theories, this one collapsed entirely when it was found that venting actually increased the crankshaft thrust several times over that resulting from the opening. Needless to say, the venting holes were removed.

As previously mentioned, our goal was high torque with low engine stall and overrun speed. That this goal was attained satisfactorily can be seen in Figs. 5 through 8. Fig. 5 shows the performance of the 12½-in. converter with constant input torque. Fig. 6 shows the performance behind the FirePower V-8 engine in a Chrysler New Yorker automobile. Note in Fig. 5 the torque ratio of 2.6/1 at stall, and in Fig. 6 the relatively low stall speed of 1490 rpm. Figs. 7 and 8 are comparable performance curves for the 11¾-in. unit, again with constant input torque, and then in a Dodge V-8 automobile. The performance characteristics of these converters are based on many considerations, and we believe they represent extremely fine compromises to fit so many models in such a satisfactory manner. Let us examine these characteristics:

1. Engine stall speed: The engine stall speed was made less than the engine speed for maximum torque, even though the performance is better when the stall speed is higher. The higher stall speed results in increased gasoline consumption, greater cooling needs, and, of course, much more engine noise. Two limits were arbitrarily set up for maximum stall speed: 1400 rpm in 6-cyl models and 1500 rpm in 8-cyl models.

2. Torque ratio: The Chrysler converters have very high torque ratios to go with 2-speed trans-

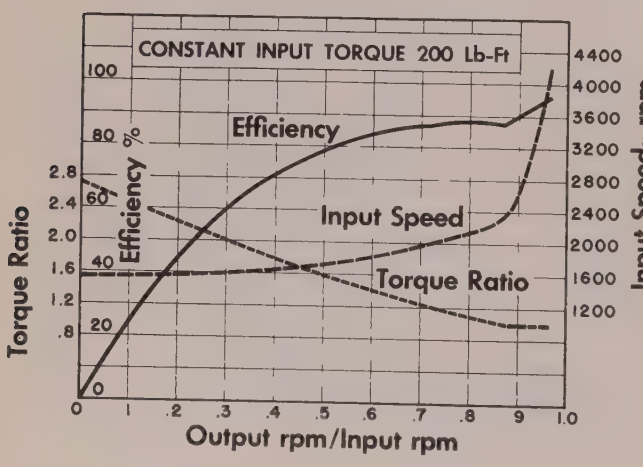


Fig. 7 – Performance of 11¾-in. torque converter

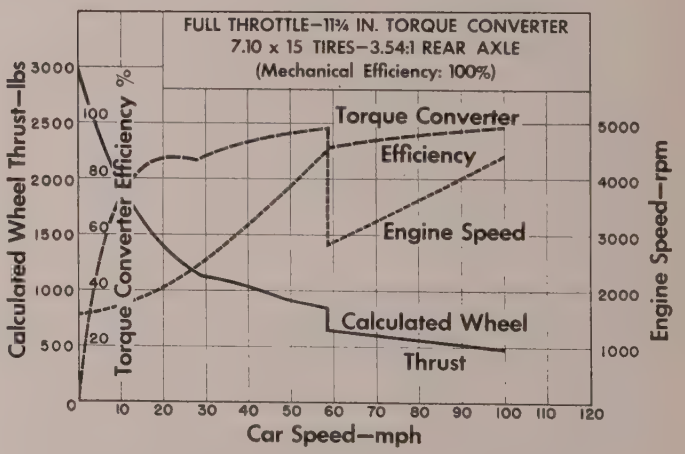


Fig. 8 – Performance of Dodge V-8 with PowerFlite

missions. Stall torque ratio is a compromise between satisfactory stall speed, peak efficiency in the torque conversion range, the speed ratio at which the converter reaches the coupling range, and the fluid coupling range efficiency. These ratios are now the highest used in passenger cars. This was achieved by careful combination of vane angles and shapes.

3. Efficiency: Efficiency in the torque converter range is more easily obtained with the 4-element unit. However, 3-element combinations have been developed with comparable conversion range efficiencies and superior fluid coupling range efficiencies.

Since a very high percentage of driving is done in the coupling range, the slip in that range is of paramount importance in achieving economy. The speed ratio at which the converter starts operating as a coupling should begin at fairly high speed ratio, say 0.87 or higher.

The use of a torque converter of generous diameter was the most important factor in securing the efficiency in the fluid coupling range. It is also our experience that 3-element converters are the most efficient because there are fewer gaps to bridge in the oil path. As a matter of fact, we are now planning to make the change to a single-stator unit in the very near future.

4. Coasting: The descent of long, steep grades received considerable attention. A torque converter has a very limited ability to drive an engine from the rear wheels, particularly at low car speeds. Our experiments on Mount Washington, which incidentally has the most difficult grade in the country, proved that a converter of ample diameter helps to achieve good coasting performance.

Converter cooling is by water in the larger V-8 installations (Chrysler and De Soto). This is done in the conventional manner with an oil-to-water heat exchanger mounted on the front of the engine block. All other model applications are aircooled. This is worthy of considerable comment. Fig. 9 shows the flow of air through a universal torque converter housing, which may be applied to 6- or 8-cyl models with either size converter. This housing is an aluminum die casting weighing 12 lb. At the rear are four screened openings for letting the air in. On the converter shell are 11 pairs of cooling fins, unequally spaced to prevent noise. These fins pull air through and discharge it at the right-hand bottom side. This arrangement is capable of handling 280 cfm of air at 1700 engine rpm.

In the development of aircooling, it was difficult to arrive at the proper requirement. It was decided that the most severe condition that should be met would be hauling a 4500-lb trailer up Newfound Gap near Gatlinburg, Tenn. These conditions were duplicated by towing tests in Detroit and confirmed on the actual grade.

This torque converter housing is comparatively

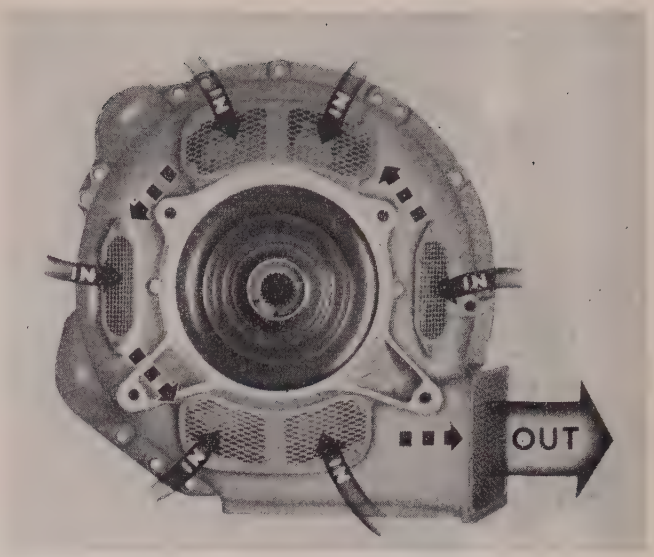


Fig. 9 – Airflow through converter housing

large and is a new application for die casting in this size. Therefore, a very extensive stress analysis and fatigue testing program was undertaken. Several modifications to bolting flanges and sections around the transmission mounting face were made to reduce stress and make more efficient use of the material.

The Planetary Gearbox

The principal components of the transmission gearbox are two planetary gearsets, one multiple-disc clutch, two bands, two pumps, nine control valves, and a governor. Reference to the cross-section in Fig. 10 will show the relative location of all these elements.

The input to the gearset is by the front annulus and rear sun gear, with the output from the rear annulus. In operation, all starts are made in low gear wherein the front sun gear is held by the front or kickdown band. This produces an underdrive ratio of 1.72/1. As the car accelerates, an

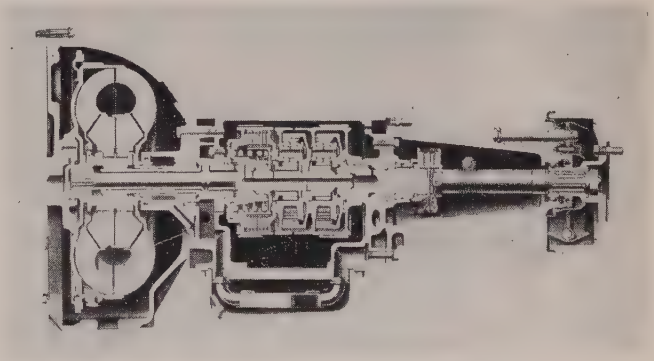
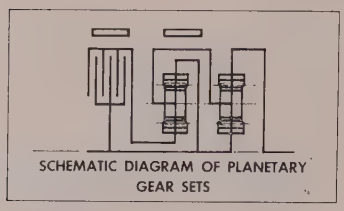


Fig. 10 – Cross-section of PowerFlite transmission



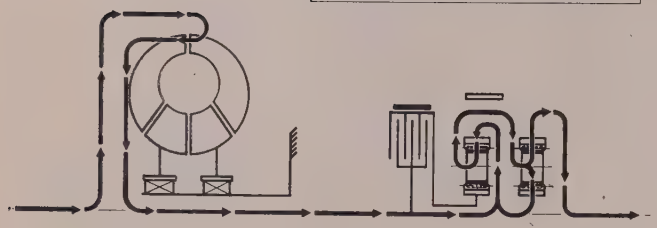


Fig. 11 – Power flow – low

automatic power upshift is made to direct by releasing the band and engaging the direct clutch, which clamps the front sun and annulus gears together. A kickdown from direct into low gear can be made by pushing the accelerator pedal to the wide-open throttle position. In low range, the controls do not allow the power shift to direct but hold the transmission in the kickdown gear. Reverse is obtained by holding both planet carriers with the rear band. This produces a ratio of 2.39/1. In neutral, all friction elements are disengaged. Figs. 11, 12, and 13 illustrate schematically the power flow through the gearbox, while Table 1 tabulates the various gear ratios.

The construction of the transmission is relatively simple. The gearset and direct clutch form a compact subassembly, which can be inserted into the case fully assembled. Exposed at the front of the transmission are the front pump and, behind it, the regulator valve body. The governor and rear pump are readily accessible at the rear by removing the extension. By lowering the oil pan, the main valve body and transfer plate are accessible and may be removed for servicing. The two band

Table 1—Transmission and Overall Ratios

	Transmission Gear Ratio	Converter Ratio	Transmission Overall	Overall 3.36 Axle
Low	1.72	2.60 at stall	4.47 at stall	15.02 to 5.78
High	1.00	1.60 at 12 mph	1.60 at 12 mph	5.38 to 3.36 at 12 mph
Reverse	2.39	2.60 at stall	6.21 at stall	20.85 to 8.03

servos are an integral part of the case, removing a potential source of leakage. The reverse band and linkage are all within the transmission and have an internal adjustment for wear. An external adjustment is provided for the low band.

Six discs splined on their inner diameters and six splined on their outer diameters form the direct clutch friction elements. The cork and Krafelt lining is placed on the discs splined on their inner diameters. The malleable-iron direct clutch cylinder is mounted on the reaction shaft extension. Sealing of the clutch aluminum piston is accomplished by an outer neoprene seal and an inner cast-iron ring. Release of the clutch is handled by a 250-lb coil spring. To prevent an unwanted engagement of the clutch due to centrifugal force at high speeds, a bleed valve in the form of a ball is placed in the clutch cylinder.

The front or low band has an asbestos friction material. It is engaged by a servo carrying both line pressure and a throttle-modulated pressure. A resin-bonded metallic friction material is bonded with Cycleweld cement to the reverse band. The reverse servo is actuated by line pressure and is cushioned hydraulically.

Both pumps are of the internal gear type. The front pump is driven by a quill to allow for some misalignment and runout of the torque converter. The quill runs in a bushing and pilots the pump pinion. All parts of the front pump are of cast iron. The rear pump, driven by the output shaft, is similar to the front except for size and materials. The

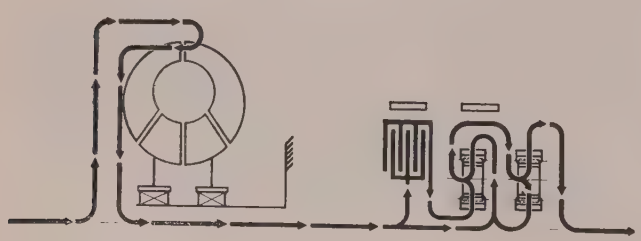


Fig. 12 – Power flow – direct

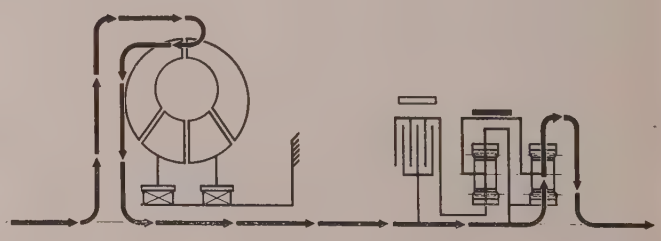


Fig. 13 – Power flow – reverse

housing is an aluminum die casting and the gears are steel. Hardening of the outer gear is beneficial to the life of the pump.

Hydraulic Controls

The PowerFlite hydraulic control system appears to be complex, as does that of any automatic transmission. Actually, two oil pumps, two bands, and a clutch are controlled by only nine sliding valves. The interrelation of the control valves with the operation of these pumping and clutching devices can be more readily understood if the function of each basic valve is clearly defined.

The six basic valves are:

1. Regulator valve.
2. Torque converter control valve.
3. Manual valve.
4. Throttle valve.
5. Governor valve.
6. Shift valve.

Fundamental to the operation of any valve is a supply of oil under pressure. Two internal gear oil pumps, one driven by the engine and the other by the transmission output shaft, deliver this oil to the regulator valve. The front pump has a theoretical capacity of 3¾ gpm at 1000 rpm and the rear pump a capacity of 2 gpm at the same speed.

1. Regulator valve: The regulator valve is designed to maintain a line pressure of 90 psi in neutral, drive, and low, and 250 psi in reverse. These pressures are affected by neither throttle position nor car speed.

When the main control system has been pressurized to 90 psi, the regulator valve directs oil to the torque converter. Further buildup in pressure causes additional movement of the regulator valve, thereby connecting front pump pressure with suction. If, because of rear pump output, pressure rises still more, the valve also directs this oil to front pump suction. The car speed at which the rear pump has assumed full control is termed "crossover" and is approximately 35 mph. This practically removes any head against which the front pump must work, thereby diminishing horsepower losses.

2. Torque converter control valve: The oil directed from the regulator valve to the torque converter must pass through the torque converter control valve as it enters the converter and through

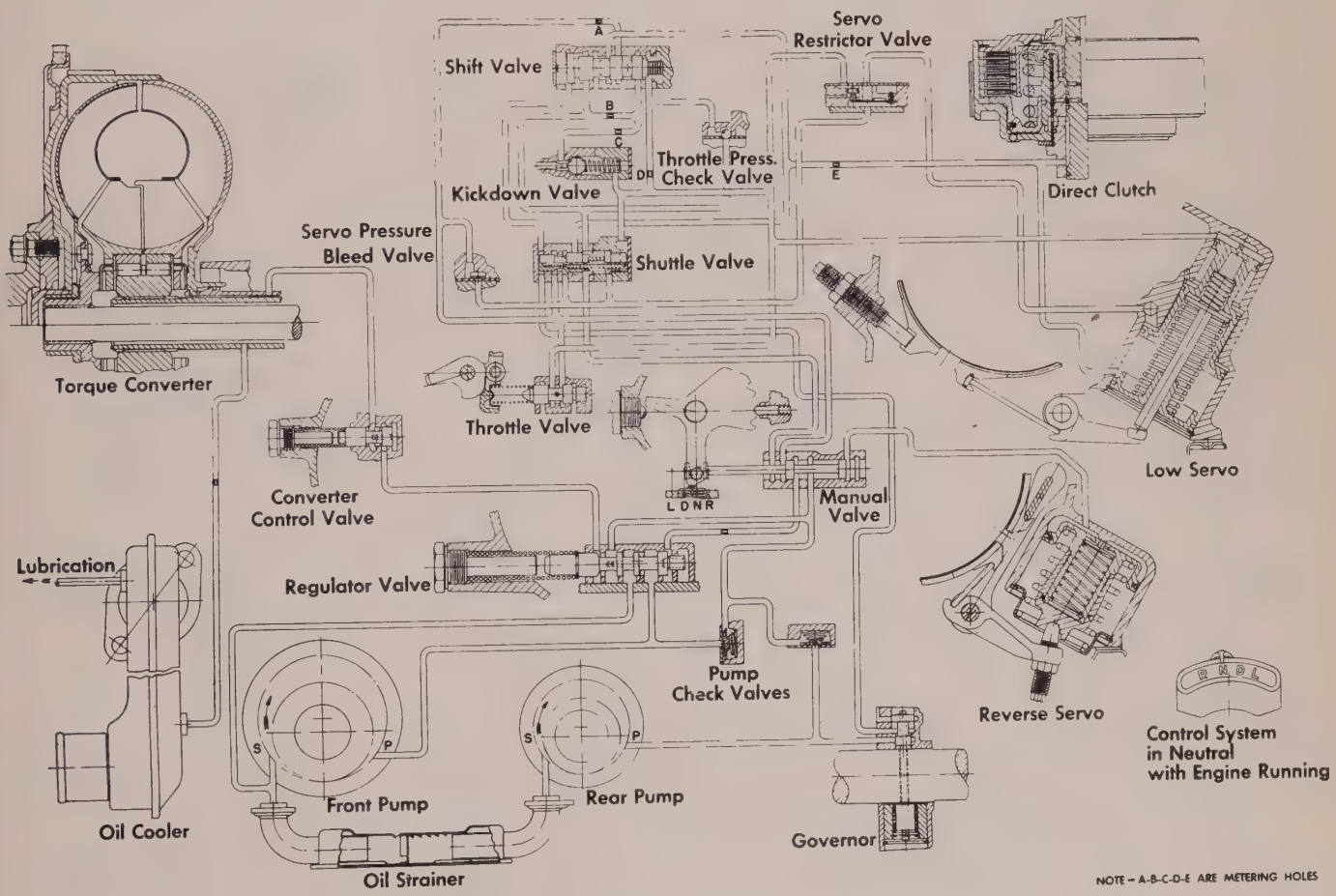


Fig. 14 – Hydraulic circuit – neutral

a metering orifice as it leaves. With increased flow, the pressure drop across this orifice increases, gradually increasing torque converter pressure. When a pressure of 60 psi has been reached, the control valve at the converter inlet starts modulating the supply of oil delivered by the regulator valve so that converter pressure never exceeds that figure.

Oil leaving the converter through the metering orifice flows to the oil cooler on water-cooled installations and then to the transmission lubrication system. On aircooled transmissions, oil flows directly from the converter outlet to the lubrication system. At normal operating temperatures, this flow amounts to about 10-12 lb per min at 25 mph. There is flow through the converter as long as any oil leaves the regulator valve, thus ensuring transmission lubrication at all times.

3. Manual valve: The manual valve directs line pressure to the valves and servos employed in the selected operating range. It also serves to increase line pressure in reverse by venting a portion of the regulator valve reaction area.

4. Throttle pressure valve: The throttle pressure valve is one of the valves pressurized in the forward driving ranges, and it sets up a pressure that is a function of throttle opening. At closed throttle the valve of this pressure is 15 psi and at wide-open throttle 90 psi. Throttle pressure serves two functions. It modulates band and clutch loads and it establishes the speed at which an upshift will occur for any given position of the accelerator pedal.

5. Governor valve: This valve produces a pressure proportional to car speed. It is used in conjunction with throttle pressure to determine the speed at which the transmission should shift from the 1.72 gear ratio into direct. During a kickdown, governor pressure is used to control the quality of the downshift by working in conjunction with the shuttle valve, to be mentioned later.

6. Shift valve: This valve, with throttle pressure acting on one end and governor pressure on the other, determines when a shift up or down should be made. It feeds oil under pressure to the proper clutching elements during an upshift and drains the oil from these clutching elements during a downshift.

These six valves are absolutely essential to the transmission operation.

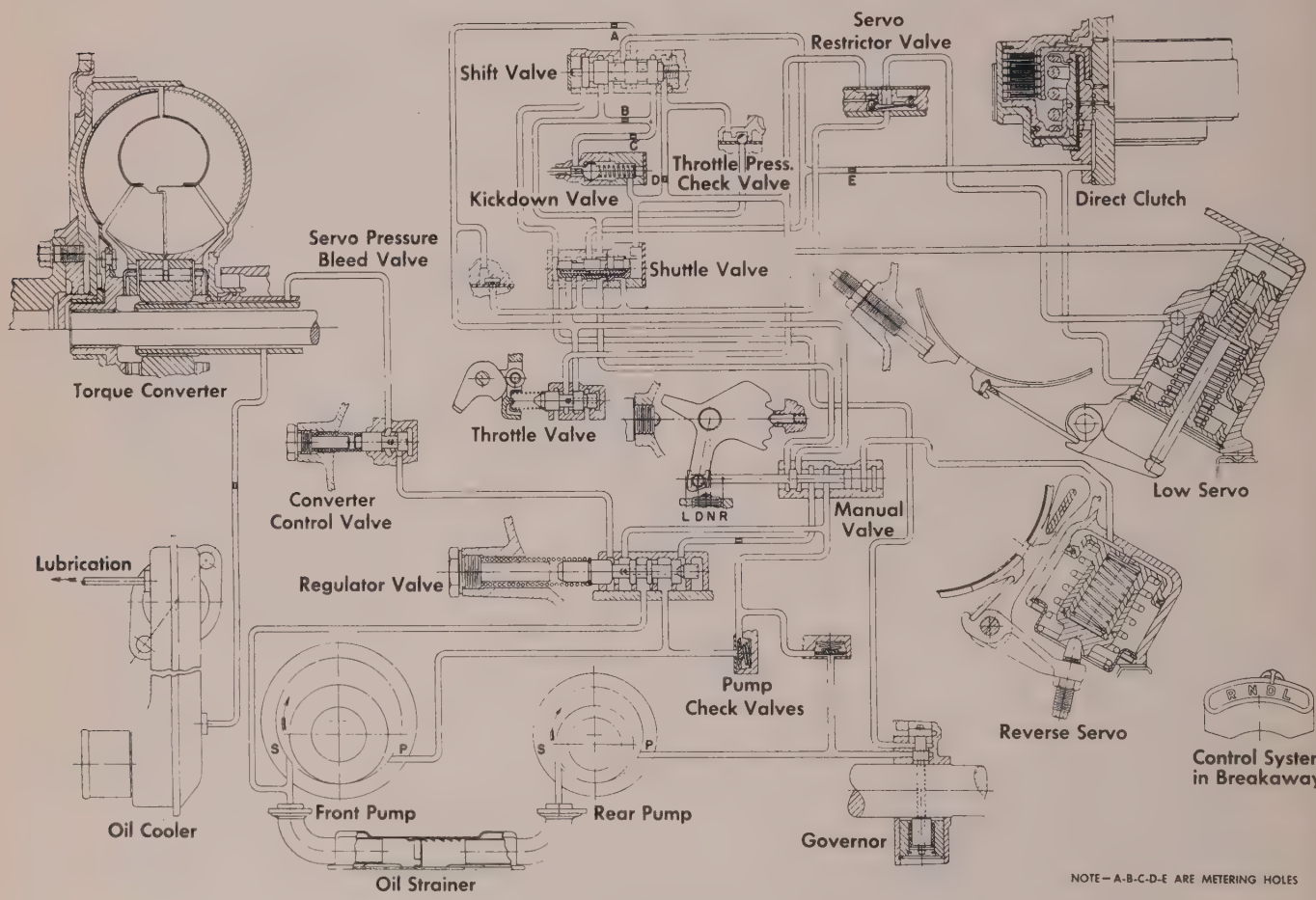


Fig. 15 – Hydraulic circuit – breakaway

Other valves used to refine operation include:

1. Shuttle valve.
2. Servo restrictor valve.
3. Kickdown valve.

A more complete description of the shift valve and of the shuttle valve will follow the discussion of general operation.

General Operation of Controls

Neutral – To start the car the selector lever is placed in the neutral ("N") position. (See Fig. 14.) Only in this position is the starting switch closed. With the engine unable to start in any other position, the danger of a sudden lurch in drive or reverse is eliminated. The manual valve is positioned in neutral to prevent any oil from reaching valves other than the main regulator valve and the converter control valve. The clutch and band servos are vented. Line pressure is regulated at 90 psi, the converter is pressurized and oil flows to the lubrication circuit. Only the front pump, driven by the engine, is operative.

Drive – Breakaway – In drive range ("D"), the manual valve admits oil to the control valves and to the kickdown band servo. (See Fig. 15.) With the kickdown band engaged, the transmission is in its 1.72 gear ratio. The throttle valve establishes a pressure proportional to throttle opening (15 to 90 psi). This pressure is applied to the right end of the shift valve and to the throttle-pressure area of the kickdown servo. As car motion begins, governor pressure rises and acts on the left end of the shift valve. An upshift occurs when the car has attained sufficient speed at a given throttle opening, and the governor pressure acting on one end of the valve is sufficient to overcome the forces of the throttle pressure and of the spring acting on the opposite end of the valve.

Direct – The shift valve (Fig. 16) now is in the extreme right position, where it directs oil pressure to the clutch. The same oil entering the clutch also enters the "off" side of the kickdown servo. As this pressure rises, the holding capacity of the band is gradually diminishing while, at the same time, the clutch holding capacity is increasing so that smooth transition from band to clutch is obtained. During the time that the kickdown servo is moving from the on to the off position, clutch capacity is

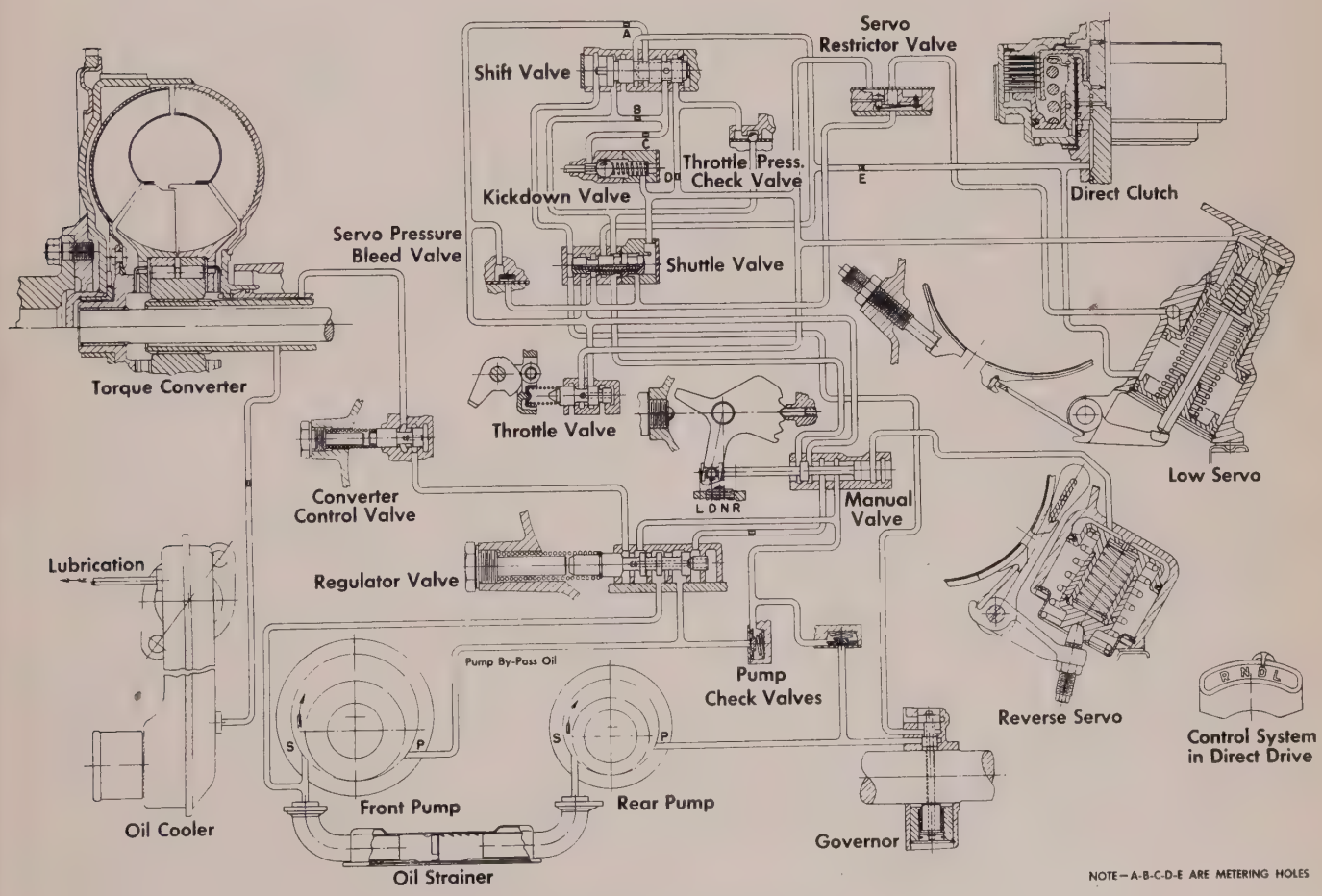


Fig. 16 – Hydraulic circuit – direct

such that a gradual engagement of the clutch is accomplished and a smooth shift is made to direct. At light throttle, the upshift occurs at 17 mph on a Chrysler V-8 with a 3.36/1 axle ratio. The same car will upshift at 65 mph at wide-open throttle. It should be noted that the pressure in the clutch *at the time of the shift* is a function of throttle opening corresponding to various engine torques. At light throttle this pressure is 50 psi and at full throttle 70 psi. This variation in clutch pressure is accomplished by changing throttle pressure acting on the kickdown servo piston.

Downshift – The normal downshift occurs at 11 mph regardless of throttle position. The action during a downshift is the opposite to that of an upshift. Governor pressure decreases with decreasing car speed until, at 11 mph, it cannot resist the force created by the spring on the opposite end of the shift valve. The shift valve then returns to the extreme left position, where it cuts off the flow of oil to the clutch and vents the clutch and "off" side of the kickdown piston. The clutch loses capacity and releases, while the kickdown band gains capacity and puts the transmission back into the 1.72 gear ratio.

Kickdown – This is merely a forced downshift. When the accelerator pedal is pressed to the floor, the shift valve moves to the downshifted position due to the hydraulic unbalance established when the kickdown valve opens. On the New Yorker, kickdown may be made between 12 and 55 mph.

Low – With the selector lever in the low ("L") position (Fig. 17), the transmission remains in kickdown gear at all times. When in direct drive, the transmission should not be manually placed in low gear above 65 mph. The mechanism of the shift consists in moving the shift valve to the extreme left by directing oil through the manual valve to the right-hand reaction areas on the shift valve.

Reverse – With the manual lever in the reverse ("R") position (Fig. 18), oil is supplied to the converter and reverse servo only. In order to handle the high band loads, reverse pressure is increased to 250 psi by venting one reaction area of the main regulator valve.

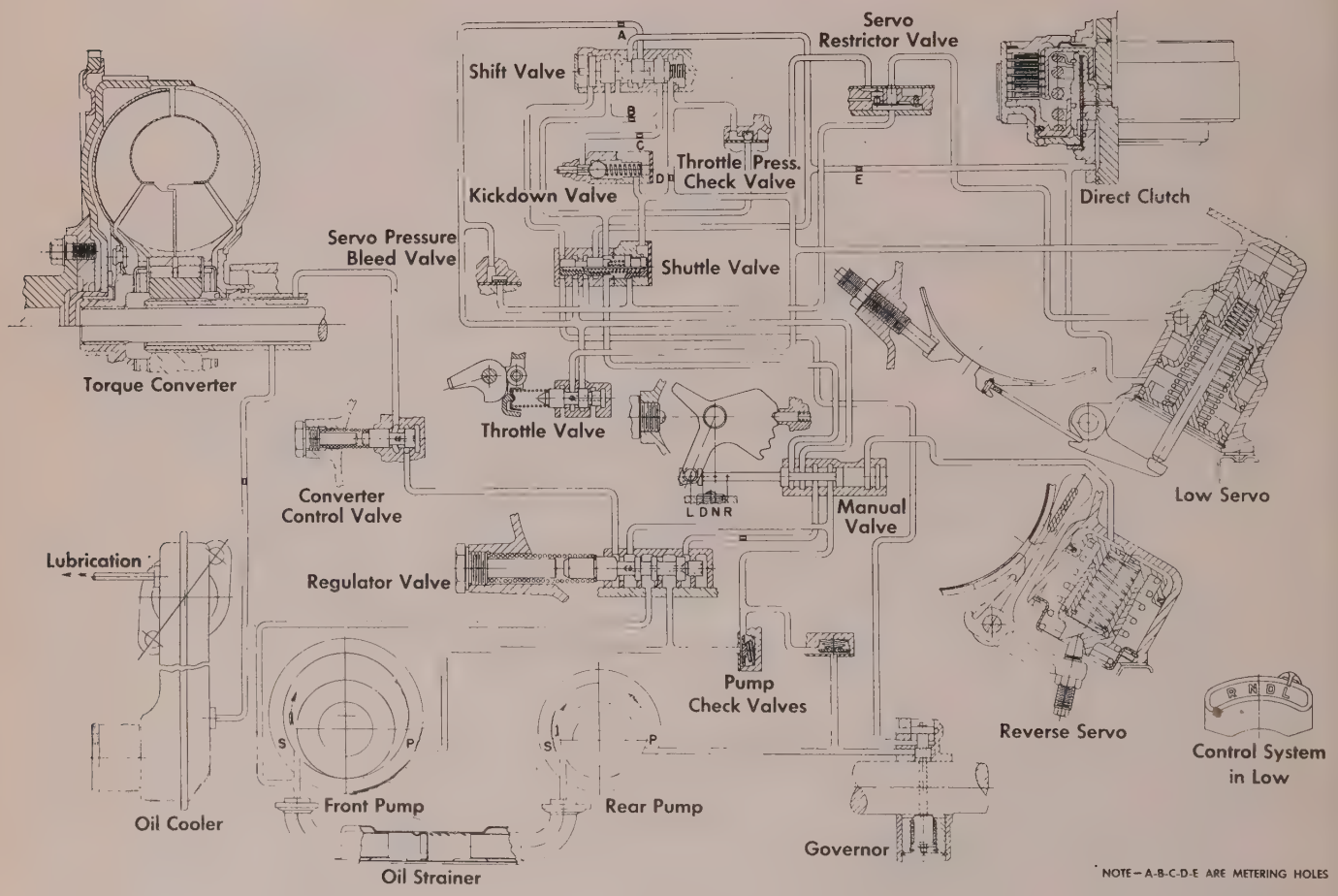


Fig. 17 – Hydraulic circuit – low

Rocking Car – Rocking the car with the PowerFlite transmission is very simple. The movement of the manual lever from reverse to low alternately applies the low and reverse bands. (As already mentioned, rocking the car can be easily accomplished by lifting the gearshift lever and moving it from one end of the indicator dial to the other. As this can be done without the necessity of watching the indicator pointer, greater safety is provided when rocking a mired car – or maneuvering into a narrow parking space.)

Push Starting – For a push start, the ignition switch is turned on with the selector lever in neutral. The car is pushed until it is moving at 25 mph. A shift into low range engages the low band and picks up the engine.

Unique Control Features

The shift valve and shuttle valve are two outstanding developments in the control system. The servos and governor also deserve some description.

Shift Valve – In any shift valve arrangement, one of the prime requisites is that, once shift valve motion begins, the valve must move rapidly to its opposite extreme of travel. A hesitation in valve motion could mean failure of one or more clutching elements because of slippage. In particular, shift valve "snap" must be under way before oil is turned on or off between the supply and the pertinent clutching elements. There are several ways in which this snap action may be accomplished. One of the methods often employed is a compensating valve in the shift valve train such that initial movement will produce an unbalance in compensated pressure and snap the valve during a shift.

In the PowerFlite transmission, the snap action of the shift valve is accomplished hydrodynamically. For clarification of the manner in which this snap occurs, assume a fluid moving in a closed tube with no restriction at the downstream end. If two similar orifices of equal area are stationed at two points, "a" and "b" in the tube, the pressure between "a" and "b" will always be half the pressure upstream from "a."

Now let us see how this principle applies to the shift valve (Fig. 19).

Let us assume that a wide-open-throttle upshift

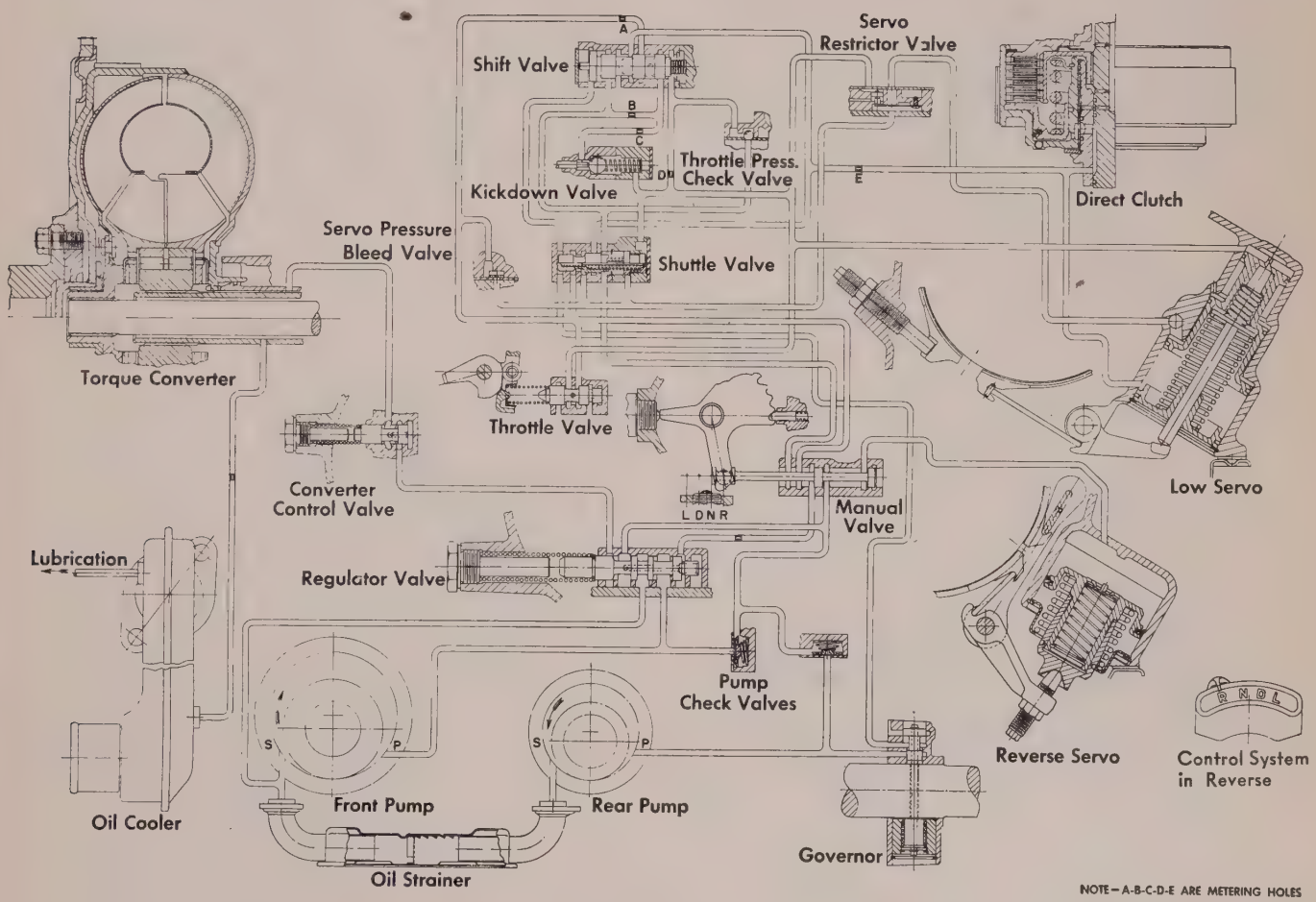


Fig. 18 – Hydraulic circuit – reverse

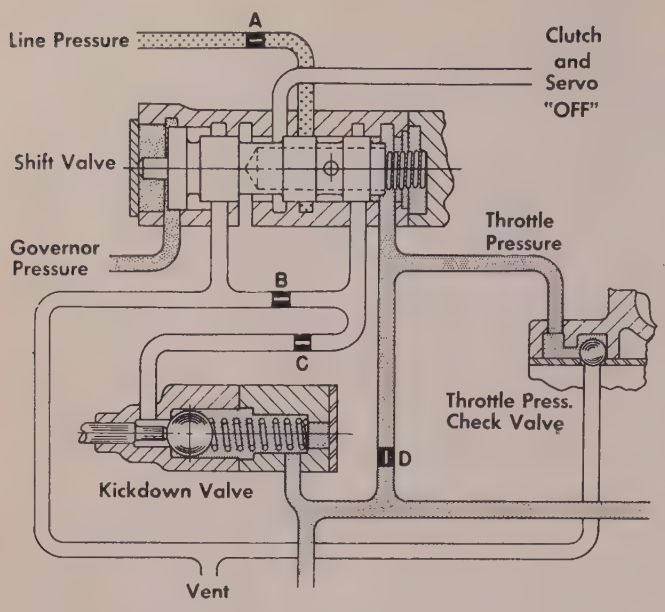


Fig. 19 – Shift valve

is about to take place. Throttle pressure upstream from orifice D is 90 psi and acts on the right end of the shift valve. At about 65 mph, governor pressure acting on the left end of the shift valve overcomes the forces of throttle pressure and spring acting on the right end. As the shift valve starts moving to the right, the vent port is just barely uncovered, allowing oil supplied through D to flow out through B. As in the straight pipe analogy, the pressure between D and B falls from 90 to 45 psi. With only 45 psi acting on the right end of the shift valve, it has reached a sudden state of unbalance and therefore snaps to the right, upshifting the transmission.

To summarize: During an upshift the shift valve throttle pressure falls to one-half of its original value and then to zero. During all downshifts except kickdown, pressure rises from zero to one-half of throttle pressure and then to full throttle pressure. Kickdowns are accomplished by forcing oil into the vent port of the throttle end of the shift valve. This action can be best studied on a complete control diagram.

Shuttle Valve – The shuttle valve (Fig. 20) has two important functions:

1. The first function is to provide for smooth upshifts when the driver makes a lift-foot upshift.
2. The second function is to ensure smoothness by regulating pressure to the kickdown piston when making a kickdown.

If the driver is accelerating the car at heavy throttle and suddenly elects to lift his foot from the accelerator pedal, the shuttle valve unloads the kickdown band and provides for rapid engagement of the clutch. The unloading of the band is accomplished by connecting the line pressure apply area of the kickdown servo with the "off" side of the same servo. This rapid shift eliminates the momentary engine braking effect that would otherwise accompany a lift-foot shift.

During a kickdown, the clutch releases and the kickdown band should take hold when engine speed has increased 1.72 times (kickdown gear ratio) its value in direct. If the band engages too soon, it will force the engine up to the new speed, resulting in a bump. If the band engages late, the engine will have overspeeded and a runaway bump will occur. For smooth kickdowns, the band must engage at the right time. The shuttle valve controls engaging time by allowing oil under *high* pressure to flow to the kickdown piston at low car speeds when engine speed-rise time is short. At high speeds when engine speed-rise time is longer, it restricts oil flow to the kickdown piston. The rate

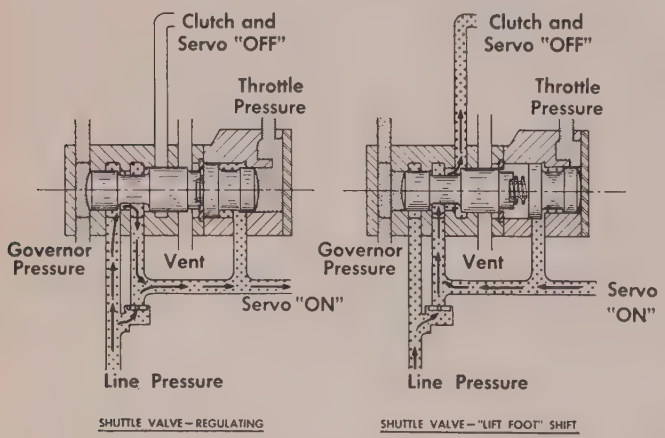


Fig. 20 – Shuttle valve

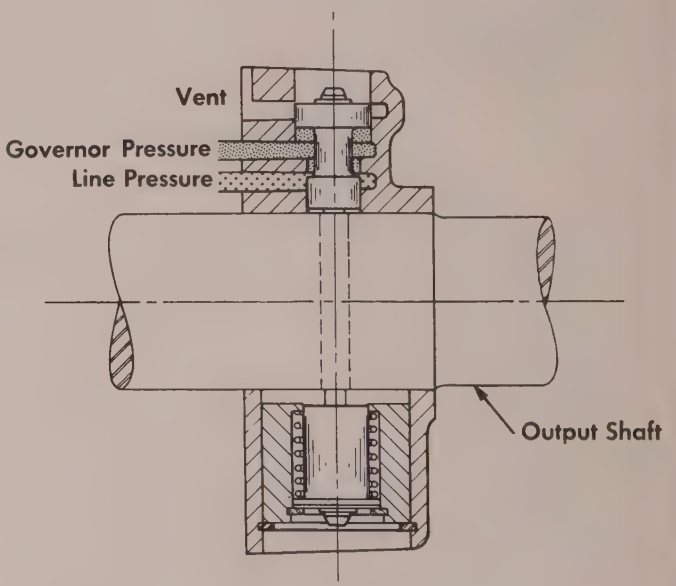


Fig. 21 – Hydraulic governor

of oil flow to the kickdown piston is modified by governor pressure acting on one of its reaction areas. The speed of band application is inversely proportional to car speed.

Governor – The governor assembly (Fig. 21) includes the governor body, valve, weights, and connecting stem. The body is mounted on the transmission output shaft, with the weights on one side of the shaft and the valve on the other. The weights and the valve are interconnected by the stem. When the output shaft is rotating, the weights are pulled away from center by centrifugal force, and hydraulic pressure acting on the governor valve through the connecting stem opposes this force. The resulting pressure curve is parabolic. To obtain the desired curve, a compound weight is used. At car speeds under 25 mph both weights act, resulting in a steep pressure-speed curve. Above 25 mph, the primary weight moves outward against the force of a preloaded spring and bottoms against a snap ring, leaving only the secondary weight to act. The resultant curve is much flatter. This compound system is used to obtain best average governor pressure curve for entire range of car speeds.

Low (Kickdown) Servo – In this servo (Fig. 22), a stepped piston applies the kickdown band through a 4/1 lever system. There are two areas on the apply side pressurized in drive and low; 90-psi line pressure acts on one and throttle pressure on the other. The purpose of these two pressures acting on the apply side is to establish a band load proportional to engine torque. During an upshift, clutch pressure directed to "off" side of the piston overcomes opposing force of line and throttle pressures and pushes piston to released position.

In neutral and reverse, the kickdown piston is held off by the kickdown piston spring. Engagement of the kickdown band is softened during the shift from neutral to drive by compression of the cushion spring.

Reverse Servo – This servo (Fig. 23) consists of a piston and sleeve arrangement actuated by oil delivered through the manual valve. Oil admitted to the piston flows into the piston sleeve, which applies the band through a 5.5/1 linkage system. Initial engagement is accomplished through the medium of oil trapped between the piston and sleeve. This trapped oil is forced out through two metering holes in the sleeve, thereby cushioning the engagement. The main piston follows and applies the full load at 250 psi.

Conclusion

The PowerFlite transmission is the result of an evolutionary program which has included 3-speed units, lockup clutches, and almost any other feature that appeared logical in the course of development. It was finally confirmed in its present form because it proved to be such an outstanding combination of the characteristics most sought in an automatic transmission. To this end, we wanted light weight, simplicity, ruggedness, dependability, performance, ease of manufacture and, certainly, a new standard in smoothness of operation. The performance of PowerFlite in our cars, and the public's demonstrated acceptance, satisfy us that, in PowerFlite, these ends have been attained.

DISCUSSION ON FOLLOWING PAGE——→

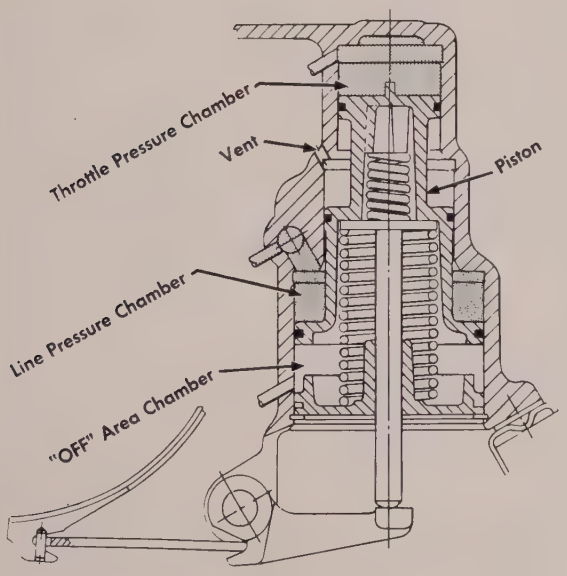


Fig. 22 – Low servo (engaged)

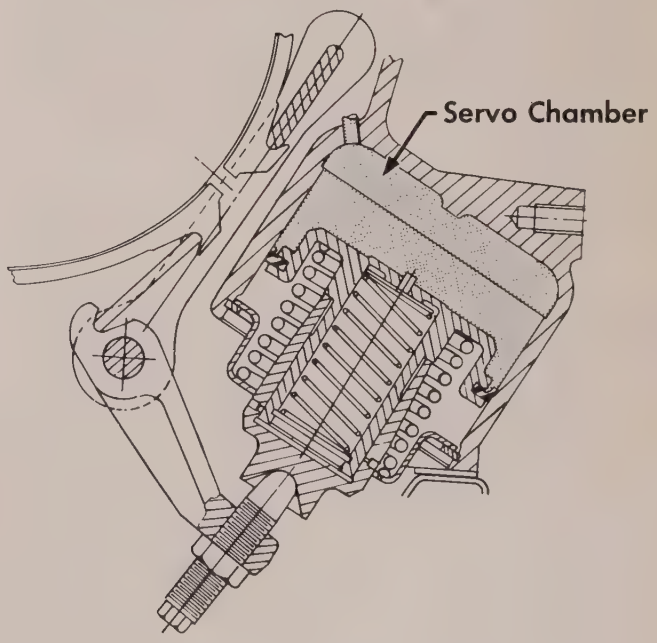


Fig. 23 – Reverse servo (engaged)

DISCUSSION

Wants More Data On Planetary Gears

– Howard W. Simpson
Consulting Engineer

I WISH the authors would have included a description and data of the planetary gears and their reasons for choosing the particular gear train shown.

The tooth loads and planet bearing loads are rather high, which apparently necessitates a fairly coarse pitch. In this respect, Chrysler is running counter to the trend toward finer and finer teeth. Finer pitches such as 16-, 18-, or 20-pitch gears have several advantages, but of course can only be used in the gear trains having light tooth loads.

Since the sliding action of gear teeth increases as contact progresses above and below the pitch line, fine teeth are inherently more efficient. They are also cheaper to cut because less metal is removed.

Since both ring gears in the PowerFlite carry 1.72 times input torque in low and both carriers are loaded at 2.44 times input torque, the tooth and bearing loads and end thrusts are higher than in a gear train in which the input ring gear carries only 100% of input torque.

It would be interesting to know how these high loads affect the mechanical efficiency in low gear. Efficiency of automatic transmission gears, in general, has, in the opinion of the writer, been sadly neglected and is all the more important because gear losses only add to a total loss that is erroneously charged to the converter by the layman. Knowing that converter losses are necessarily high is all the more reason for holding gearbox losses to a minimum to keep heat generation to a minimum.

Losses in automatics occur in the following, all but the first being in the gearbox: converter, gears, gear thrust washers, front pump, rear pump, clutch drag, brake band drag, brake drum bearings, planet bearings, and oil seals.

It is estimated that the losses in the gearbox exceed those in the converter after the latter has passed the coupling point, and it is, therefore, important to keep each loss as low as possible, so that extreme conditions such as carrying heavy loads in mountain regions can be met. In this case, excessive gear losses could be the straw that breaks the camel's back.

Automatic Transmissions Need Better Performance

– H. G. English
Ford Motor Co.

ALL of the automatic transmissions introduced since World War II have been of the torque converter type, leaving the Hydra-Matic to defend alone the position of the fluid coupling step-gear type of transmission. The converter transmissions have been of the 2-speed variety, with and without lockup clutch, and the 3-speed type also with and without lockup clutch. The most outstanding characteristic of the converter transmissions in general has been and still is their relative smoothness. Another outstanding characteristic of those that drive at all times through the converter, that is without lockup clutch, is great flexibility in the 20-40-mph range. This affords a real performance advantage in city driving.

The earliest converter transmissions were of the 2-speed type and went all the way to provide the ultimate in smoothness, even at the expense of performance, by utilizing the converter alone for all normal driving. The 3-speed designs later introduced provided more of a balance of performance and smoothness of operation by supplementing the torque converter with its intermediate gear ratio of approximately 1.48/1. This, of course, introduced one automatic shift in the drive range but provided additional starting torque and a highway passing gear.

In the past year the 2-speed converter transmissions have undergone significant changes to improve starting torque by use of higher torque ratio converters or automatically starting in low gear. The Chrysler PowerFlite transmission combines a comparatively high torque ratio converter with an automatic low gear to achieve high starting torque. It is assumed that the selection of the 1.72 low-gear ratio in contrast with the 1.82 of the other 2-speed converter transmissions was to extend the useful speed range of low gear without too great a loss in low-speed engine braking so necessary for car speed control on long downhill grades.

These recent developments appear to indicate a growing requirement for increased performance throughout the speed range. At the same time it is quite apparent that smoothness of operation must not be sacrificed to any degree.

Also interesting, I believe, is the apparent trend toward larger and simpler torque converters in the interest of a better balance of torque ratio, efficiency in the converter range, efficiency as a fluid coupling, and effectiveness of engine braking. Chrysler's plans to adopt a single stator in the near future is an indication that a favorable balance of important characteristics is still possible with comparatively high stall torque ratios.

Chrysler's extensive use of aluminum die castings in the interest of reduced weight and presumably lower cost is outstanding, and I am certain that we can expect still greater application of precision aluminum castings in the future. The majority of these applications are not easily arrived at but require considerable development to attain the required strength with a minimum of metal.

The PowerFlite selector pattern with reverse and forward positions separated by neutral is of interest in that Chrysler becomes the second major automobile manufacturer to depart from the pattern established by Hydra-Matic in 1939. As pointed out by the authors, this selector arrangement affords real advantages in terms of safety and convenience.

It is interesting to note that band servos of the PowerFlite transmission are cast integral with the main case. This design results in a rugged case construction and fewer parts through elimination of servo bodies and attaching parts. On the other hand, this arrangement would appear to afford less flexibility in adapting one basic transmission to a series of engines or to future changes in engine output characteristics. The use of high operating pressures for servo application would also appear to present a problem in maintaining sufficiently sound case castings to prevent external leakage of oil through the common case and servo walls. Perhaps the authors would care to comment on these questions regarding the integral case and servo casting.

Prefers Reverse and Low Adjacent for Car Rocking

– Forest McFarland
Packard Motor Car Co.

THE overall design of the Chrysler automatic transmission is clean and has been reduced to bare essentials. The control design in general follows well-established principles, but in detail the design shows Chrysler's solution to the problem present in upshifting and downshifting under all conditions.

In regard, first, to the steering column shift or range selection, Chrysler has chosen the same shifter lever gating as maker C in the Chrysler illustration except for the deletion of the parking position. This results in American automatic transmissions having two general types of shift gating patterns; first, those falling in the B and C groups, which are substantially the same except that those under C have the parking position added; and, second, those transmissions falling under the D group, which would include Chrysler with the parking position deleted. Many of us had hoped that the last two manufacturers would go along with the pattern previously adopted by the rest of the industry. It is our feeling that maneuvering or rocking the car can be accomplished easiest and best with reverse and low adjacent as in the first groups. We have had no service complaints, to my knowledge, with this arrangement. Reverse can be made inoperative above any predetermined car speed, if desired.

We must agree with the authors that deletion of the parking position does simplify the gating pattern. Many drivers have told us, however, that they use the parking position to provide a quick and positive lock against car movement in preference to the emergency hand brake. These reports have confirmed our original opinion that the parking position is a very desirable feature for both convenience and safety. To answer the question of ruggedness of our design, our engineers thoroughly bent up both front and rear bumpers of cars by driving into a car with the lever in the parking position and sliding the car several feet without damaging the "park" mechanism.

Differences of opinion such as just outlined, I have been told, are the reasons why we still have differences in automobiles, as well as horse races.

The selection of a 1.72 ratio by Chrysler for the geared drive as the best compromise between a performance gear when starting the car, kickdown gear at highway speeds, and a hill descent gear is interesting. We feel the final answer on the arrangement of the torque converter and gearing for best overall performance for all conditions has not been obtained to date in a design commercially acceptable. We believe this comment can also apply to the automatic stepped-gear transmission. This is no criticism of the units produced to date as Ward's Reports show that since 1940 there have been over 3,800,000 stepped-gear-type automatic transmissions built and since 1948 over 3,400,000 torque converter transmissions built, so the public must like them.

The gearing arrangement adopted by Chrysler appears to be a very straightforward way of obtaining the desired ratios. It would be interesting to know the pitch, pressure, and helix angle of the gears, also if modification is used.

Chrysler's method of welding the converter assembly and attaching directly to the flywheel certainly simplifies this construction. I am surprised that this type of mounting gave them the best result for crankshaft vibration in bending, in view of previous experience in like cases where a flexible disc with a flywheel or converter mounted on fore and aft bearings has appeared to be the best solution to this problem.

The authors refer to spin test speeds. Is this speed for a dry or filled converter? I would expect this unit to be good for speeds above this value dry. We have not heard of anyone having the courage or hardihood to spin a converter filled with even water above 6000. We have resorted to the use of a special seed to stimulate oil for speeds above this value.

The use of aluminum in the bell housing is a definite advance in weight reduction. Any comments on how stresses were reduced in this part would be of interest, also the overall weight of the transmission.

Selection of Mount Washington for hill test work was of interest to us as a survey of per cent and length of grades in the United States plotted by our engineers with these values as ordinates and abscissas, respectively, showed Mount Washington to be one of two grades that stood well above the general boundary of the pattern.

Control diagrams of any new automatic transmission are of intense interest to those who have struggled with the same problems. We always want to see how the other fellow finally did the job. I use the word "finally" advisedly. The most interesting features in the Chrysler controls to us are the shift valve, the shuttle valve, the 2-stage governor, and the reverse servo. Proper timing of the low high shift up and down under all conditions of throttle, car speed, and temperature, together with the variations in manufacturing clearances and change in frictional characteristics after usage, is a difficult problem.

The effect of the shuttle valve in tailoring the delay in engagement to suit the speed rise of the engine necessary to secure engagement without torque reversal shock is very interesting.

The 2-weight governor arrangement is an ingenious means of securing a steep slope in the curve of one governor, where it is needed particularly on high to low down shift on deceleration to establish this point within reasonable limits. I would appreciate any comments by the authors on friction lag present in this type of governor.

The quick-acting reverse piston is of particular interest to us in view of our own experience with a device of this type in our low and reverse brakes. This type of arrangement materially reduces engagement time, making it very helpful in rocking a car.

In summary, we can say, from our personal driving experience in the Chrysler New Yorker, that we would agree with the authors that this car has a smooth shifting unit with excellent performance, achieved without doubt at a minimum of cost, complexity, and weight.

Considers Engine Noisy in Direct Drive Above 40 Mph

– David T. Sicklesteel
Borg-Warner Corp.

CHRYSLER have been working on the PowerFlite automatic transmission since 1938, and they pioneered the fluid drive for passenger cars, using a fluid coupling with the standard 3-speed countershaft type of manually shifted transmission.

Next they brought out the semi-automatic 2-range, 4-speed countershaft type of transmission with a fluid drive coupling in 1941. This transmission became very popular with Chrysler drivers, and 10 years later—in 1951—was superseded with the torque converter, and now—in 1954—the PowerFlite automatic transmission is being featured.

We are happy to see Chrysler go into production on the automatic transmission similar to the automatic transmissions now in production for the automotive industry. If they had come out with something radically different, the automatic transmission engineers would now be burning more midnight oil.

I have been driving the Chrysler Windsor equipped with a PowerFlite, and find that the transmission, in general, operates very pleasantly. The torque converter is exceptionally quiet, even at stall. However, there is an objectionable amount of engine noise in direct drive above 40 mph, due, apparently, to the 3.9/1 rear axle ratio necessitated by the transmission 1.72/1 low gear ratio.

The transmission reverse ratio of 2.39, together with the torque converter ratio of 2.6, surely makes the car lively in reverse, especially when maneuvering.

Indications are that considerable thought has been given to the minute details of the hydraulic circuit to produce smooth shifting at all times. However, under certain driving conditions, noticeable jerks and bumps are still possible.

Effect of Shot-Peening Variables and Residual Stresses on the

DURING the past few years, the use of shot peening in industry has increased considerably. The list of applications of this cold-working method has grown to such an extent that it encompasses such widely diversified fields as fatigue durability, peen forming (aircraft wing panels), and control of surface roughness for prevention of galling. Yet, despite this steady growth, much of the technical information necessary for proper engineering applications of the process has not been obtained.

With the hope of shedding some light on the effect of the variables of shot peening on fatigue life and on the part the resulting residual stresses play, a program of testing and analysis was undertaken by the engineering mechanics department of General Motors Research Laboratories. The results of these investigations are presented in this paper. A simple and relatively inexpensive specimen was selected in the form of a leaf spring. These specimens were shot peened in various ways and fatigue tested in groups of four. Although from a statistical point of view much larger groups would be desired, we were primarily concerned with large significant differences which should be detectable with four specimens in each group. In all, well over 100 groups of spring specimens were heat-treated, cold worked, and tested. A few of these (nearly 50 springs) were analyzed for residual stresses. The influence of shot size, shot velocity, exposure time, and peening while specimen is under strain were

RESULTS of an investigation into the effect of shot-peening variables and the resulting residual stresses on fatigue life are reported in this paper. Leaf springs were the simple specimens heat-treated, cold worked, and tested in this study.

Some of the conclusions reached are:

1. There is a minimum shot velocity for each shot size to obtain best fatigue life, and this value is much lower than that normally used.

2. Exposure time for this type of shot-peened specimen beyond some minimum value is wasteful and costly.

3. Shot size has little influence on fatigue life for these specimens.

4. Shot peening specimens while under tensile strain greatly increases fatigue life at 200,000 psi nominal stress over that of nonpeened or strain-free-peened specimens.

5. Shot peening these specimens gave residual compressive stresses 50% of yield strength, and these stresses can be increased to more than 50% by strain peening.

6. There is a direct correlation between surface residual compressive stress and fatigue life of these specimens.

The Authors

R. L. MATTSON (M '42) is assistant head of the Engineering Mechanics Department, Research Laboratories Division, General Motors Corp. He received a mechanical engineering degree in 1940 from Michigan College of Mining and Technology and started with GMC the same year. Mr. Mattson has held his present position since 1946.

W. S. COLEMAN, JR. is a senior research engineer in the Engineering Mechanics Department, Research Laboratories Division, General Motors Corp. He graduated from Michigan State College and, in 1946, received an M.S. degree in mechanical engineering from Massachusetts Institute of Technology. Mr. Coleman joined GMC in 1946.

Fatigue Life of Leaf-Spring Specimens

R. L. Mattson and W. S. Coleman, Jr.,
Research Laboratories Division, GMC

This paper was presented before Division XX of the SAE Iron and Steel Technical Committee, Hershey, Pa., Oct. 1, 1953.

studied. Residual stress analyses were made by the dissection method, showing the influence of various cold-working treatments.

Summary of Results

The following conclusions can be drawn from this work:

1. For shot-peened specimens of this type it appears that there is a minimum shot velocity for each shot size to obtain best fatigue life. Air pressures greater than this value do not add to fatigue durability, hence they are wasteful and costly. This minimum value is considerably lower than that normally used.

2. For shot-peened specimens of this type, unless the degree of coverage (exposure time) is up to some minimum value, best fatigue life is not obtained. By using coverage greater than this value, no significant increase in fatigue life results. Hence, exposure times beyond this point are wasteful and costly. Full coverage as determined by the "knee" of Almen strip curve is adequate.

3. For shot-peened specimens of this type the influence of shot size on fatigue life is not particularly great over the full range of air pressures and exposure times.

4. For these specimens fatigue life at 200,000 psi nominal stress (approximately 95% of yield strength) was increased by 20 times over non-peened specimens.

5. By shot peening specimens while under tensile strain,[1] fatigue life at 200,000 psi nominal stress was increased by as much as 200 times over non-peened specimens and 10 times over strain-free shot peening. Greatest benefits are derived from shot peening while under strains greater than 80% of the yield strength.

6. Shot peening specimens of this type introduces residual compressive stresses of the order of 50% of the yield strength of the material, regardless of shot size and velocity. The depth of the compressed layer increases with both shot size and velocity, ranging between 0.006 and 0.026 for the treatments used.

7. Residual compressive stress considerably greater than 50% of the yield point can be introduced by shot peening while the surface is under tensile strain.

8. Although data are limited, the magnitude of residual compressive stress appears to be reduced by fatigue testing.

9. There is a direct correlation between the magnitude of residual compressive stress near the surface and fatigue life of these specimens.

Specimens

The specimens selected for this investigation were machined from hot-rolled leaf-spring stock, 0.220 x 1¾ in., of SAE 5147 material (with the exception of a few springs from 5150). The specimens were prepared as follows:

1. Cut to 12-in. lengths.
2. Numbered.

[1] See Patent No. 2,608,752.

Fig. 1 – Specimen and gage for curvature measurement

3. Straightened and shaped to approximate width.

4. Finish ground on flat surface and edges to 0.192 x 1½ x 12 in. The lengthwise edges were rounded to 1/16 radius to reduce stress concentration.

5. Heated free at 1600 F for 25 min, 2-4% excess oxygen (except where otherwise indicated).

6. Placed in fixture to produce longitudinal curvature (14 in. radius) and quenched individually in fixture in S-9 oil at 110-120 F (except where otherwise indicated).

7. Tempered at 725-760 F for 1 hr and 10 min to hardness Rockwell C 48. Tempering was done with specimens free followed by air cooling (except where otherwise indicated).

The specimens were measured for hardness and thickness and then measured for curvature using a curvature gage shown by Fig. 1. Gage length was 2.83 in., and dial indicator read to 0.001.

The specimens were then subjected to cold-working treatments, fatigue testing, or residual stress analysis, or both.

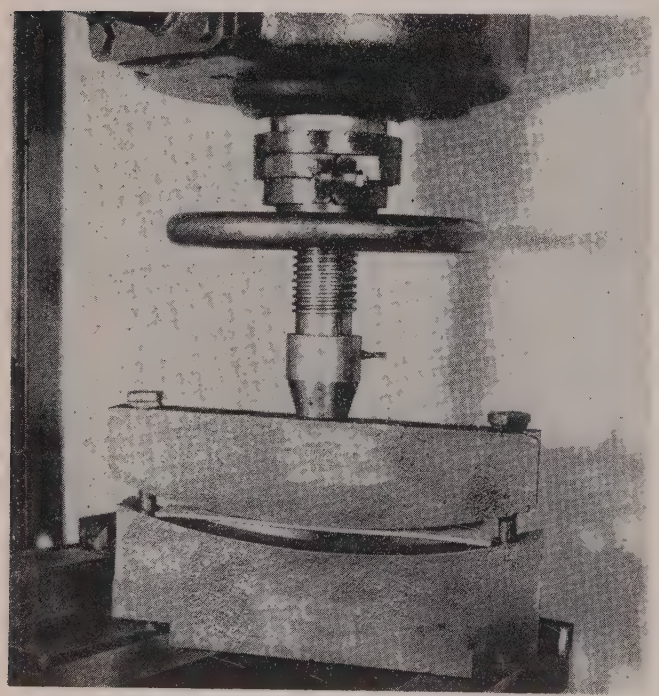


Fig. 2 – Presetting leaf-spring specimens

Presetting and Shot-Peening Procedures

The presetting operation shown in Fig. 2 consisted of placing the specimen between two blocks of predetermined curvature and loading to 5000 lb in a test machine. This load was found to be sufficient to press the specimen completely into the contour of the block. The load was maintained for approximately 30 sec and then released. A set of the preset forms was available for various degrees of preset surface strain. Presetting was always done in the direction of the applied load during fatigue testing.

All the leaf-spring specimens reported in the fatigue-results section were preset to a nominal 220,000 psi, or 10% over the test stress of 200,000 psi, prior to testing. Several of the specimens reported in the residual-stress-analysis section, however, had higher values of preset, and a few were not given presetting treatment.

The specimens were shot peened in an experimental universal air machine designed at our laboratories. As can be seen in Fig. 3, the specimens were clamped to a fixture which in turn was rotated and reciprocated beneath the air nozzle. Fig. 4 shows the fixture which will accommodate up to five springs plus three Almen strips for exposure and intensity data. Several sets of forms are available to clamp the springs in various states of surface tension for strain peening.

Shot used was commercially available chilled iron shot screened accurately to size. The size distribution was better than the present SAE size standards.

Shot flow rate was controlled by an orifice.

Fatigue Tests

1. *Test Method* – The method used for fatigue testing these specimens is one developed by the Structure and Suspension Development Department of Central Engineering Staff, General Motors Corp. Basically, this is a deflection-type test in which the specimen is bent cyclically from nearly zero stress to a predetermined curvature. This is accomplished on the stroking machine shown by Fig. 5. The specimen is placed on two stationary

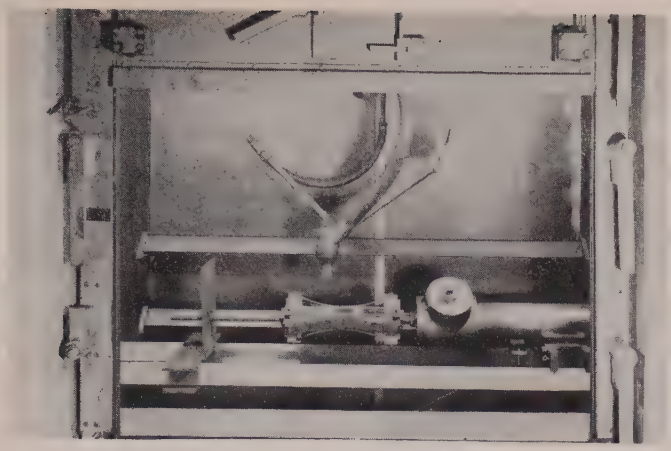


Fig. 3 – Shot-peening setup

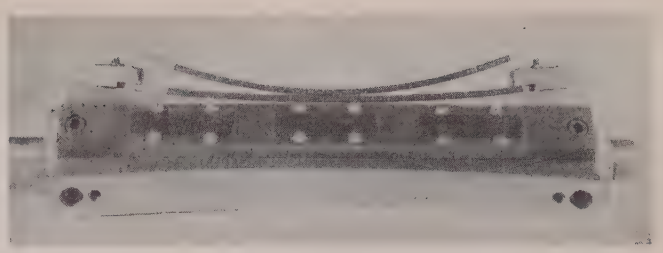


Fig. 4 – Fixture for peening leaf-spring specimen under stress. Note specimen with identical curvature in free position on top of clamped specimen

support points and loaded by forcing a "shoe" of selected curvature against the top of the specimen. The "shoe" was fastened to an arm which was pivoted at one end and connected to a connecting rod at the other. The connecting rod was attached to an adjustable eccentric on a shaft driven by an electric motor. As the shaft rotated, the shoe oscillated up and down, cyclically loading the specimen. Adjustment for length of stroke was made by adjusting eccentricity. Position of the stroke was adjusted by lengthening or shortening the connecting rod by means of a turnbuckle arrangement. For testing the specimens, the stroke and position were adjusted so that at maximum deflection, the center 6 in. of the specimen was in contact with the shoe. This constituted the test zone of the specimens. At the low-load end of the stroke, the adjustments were made so as to load just barely the specimen. An automatic shutoff was provided by means of a microswitch which detected whether or not the spring was seated against the shoe. When the spring failed, it no longer maintained close contact with the shoe, thus actuating the microswitch. A mechanical counter served to count the strokes or cycles. Loads were not measured. The basis for the test was simple curvature change. The average longitudinal curvature of the specimens in the 6-in. test length was first measured and a shoe then selected so as to cause a curvature change corresponding to a stress of 200,000 psi at the surface.

2. *Results and Discussion* – The results of the fatigue tests are shown by Figs. 6-13, inclusive.

(a) Primary Data – Fig. 6 gives the fatigue data for shot peening with 130 shot as compared with nonpeened specimens and sandblasted specimens. As a baseline, nonpeened springs had a fatigue life of 4500 to 5900 cycles, as can be seen by Batch 28.

The mildest treatment in which we were interested at the time was sandblasting which gave a substantial improvement in fatigue durability over nonpeened specimens. Fatigue life of 22,000 to 74,000 cycles was obtained as compared with 5000 for nonpeened. The only effect of increasing the time of exposure to the blast by a factor of 4 was an apparent increase in scatter.

The next treatment shown by Batch 70D is for a mild shot-peening treatment. The shot was rather small, being about 0.013 in. diameter; but the air pressure, 50 psi, was quite high so the shot velocity must have been quite considerable. A gain in fatigue durability can be noted – something of the order of 20 times the life of a nonpeened spring. To substantiate these results, further duplicate tests were run at different times. The results are shown by 44D and 68D, and also for the same mate-

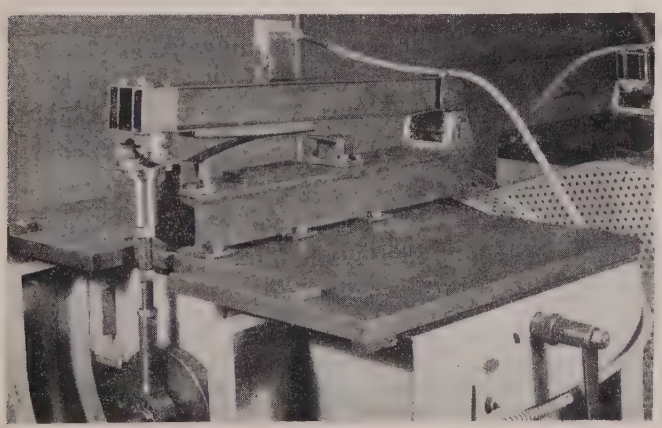
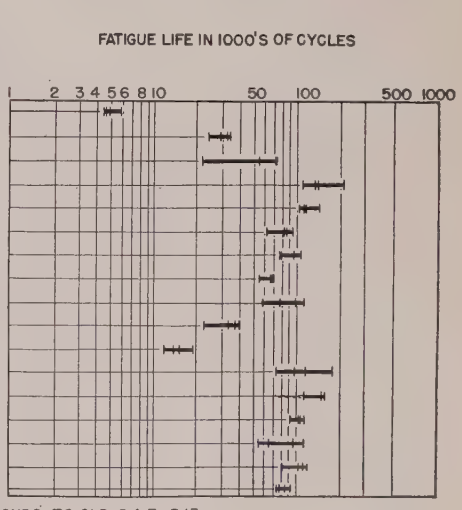


Fig. 5 – Stroking-type machine for testing leaf-spring fatigue specimens

SHOT PEENING TREATMENT

BATCH NO.	SHOT SIZE	AIR PRESS PSI	TIME EXP MIN.
28D	NO PEENING		
37D	SAND	80	1
31D	SAND	80	4
70D	*130	50	4
68D	130	50	4
44D	130	50	5
113N	130	50	5
93N	130	50	5
55D	130	75	5
54D	130	25	5
43D	130	50	0.5
41D	130	50	1
39D	130	50	2
33D	130	50	5
53D	130	50	10
40D	130	50	15
42D	130	50	30



*CORRESPONDS TO OLD S.A.E. P13

Fig. 6 – Fatigue-test results for 130 shot

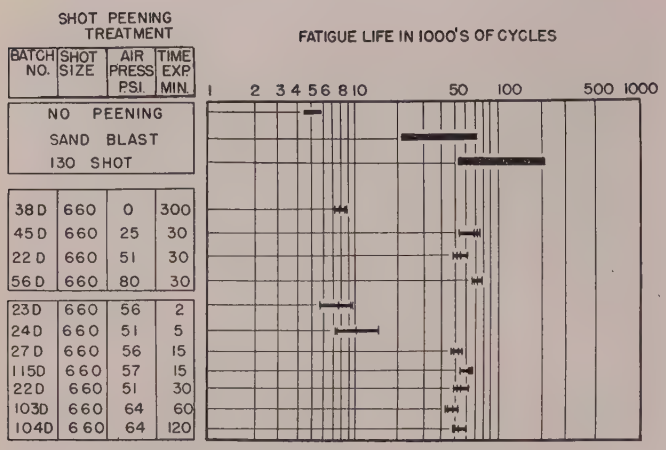


Fig. 7 – Fatigue-test results for 660 shot

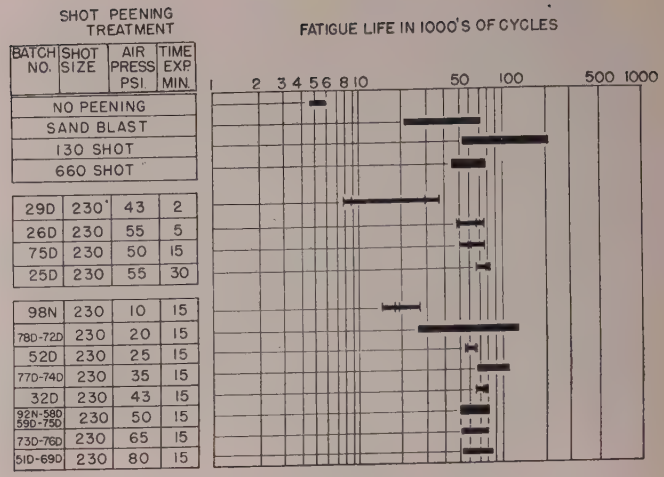


Fig. 8 – Fatigue-test results for 230 shot

rial, same heat-treatment but from a different heat of steel, 93N and 113N (the suffix designated the heat of steel).

Shown also are data for various air pressures and exposure times which will be discussed later.

Next, the effect of shot size was examined. The largest present commercially practical size was SAE 660; hence, data on this size provides information pertaining to the upper limit of this variable. Tests were conducted varying air pressure (for shot velocity) and exposure time (for coverage). The results are shown by Fig. 7 which includes the previous test data in an abbreviated form at the top.

Tests were also made using SAE 230 chilled iron shot varying air pressure and exposure time. The results are shown by Fig. 8. For reference, again, the previous data are compressed into the upper part of the diagram.

The previous data are summarized in Figs. 9-11 inclusive.

(b) The Effect of Exposure Time – The influence of exposure time is shown by Fig. 9.

The vertical scale is time of exposure, the horizontal scale to the right is life in cycles, and the horizontal scale to the left is arc height of an Almen C strip.

First of all, it can be seen that exposure times must reach some point before maximum fatigue durability is obtained. Furthermore, continued exposure beyond this point is of no particular benefit and of no particular harm for these specimens. This particular magnitude of time of exposure appears to be less than that corresponding with the "knee" of the intensity curve. This is detectable best from the curves for SAE 230 and SAE 660 shot. The arc height continues to rise even after maximum benefits in fatigue durability have been attained. Coverage obtained with an exposure time corresponding with the "knee" therefore is generous. Some practical means are needed for selecting this critical degree of coverage. This diagram shows that for these specimens, somewhat greater fatigue life was obtained with SAE 130 shot than with either SAE 230 or SAE 660 and that slightly better fatigue durability was obtained with SAE 230 than with SAE 660.

(c) Effect of Shot Velocity – Shot velocity was not actually measured nor was it computed, but it was taken to be related in some simple way with air pressure. Hence, Fig. 10 shows air pressure versus fatigue life for various shot sizes.

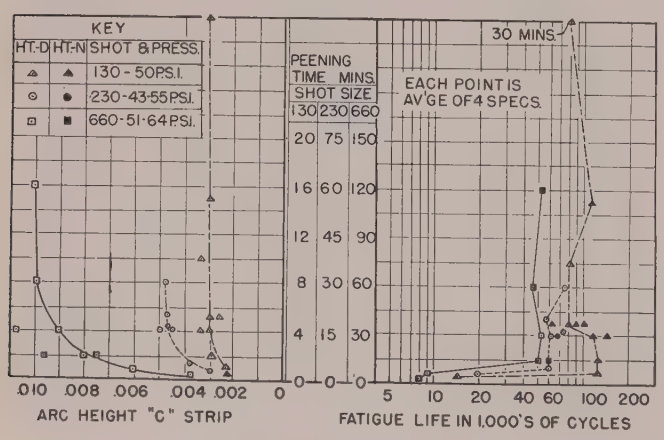


Fig. 9 – Effect of peening time on fatigue life

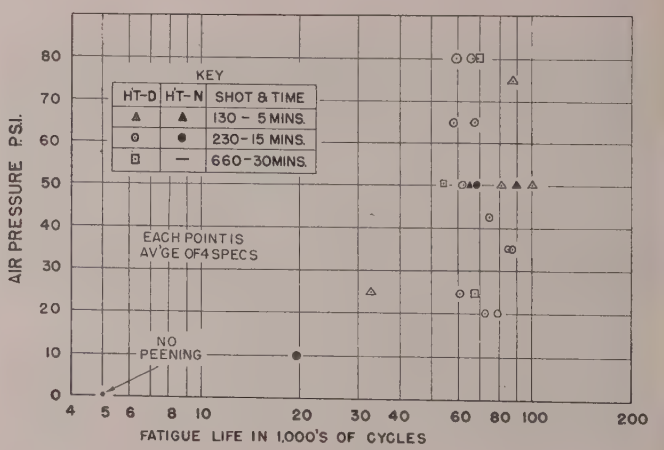

AIR PRESSURE P.S.I.
80 70 60 50 40 30 20 10 0
KEY
HT-D HT-N SHOT & TIME
130 - 5 MINS.
230 - 15 MINS.
660 - 30 MINS.
EACH POINT IS AV'GE OF 4 SPECS
NO PEENING
4 5 6 8 10 20 30 40 60 80 100 200
FATIGUE LIFE IN 1,000'S OF CYCLES

Fig. 10 – Effect of air pressure on fatigue life

The exposure times were constant for each shot size.

From these data, it appears plausible that for each shot size there is a minimum air pressure required to obtain maximum fatigue durability and that beyond this value of air pressure, no further gain in fatigue durability is obtained. This value of air pressure is less for the larger sizes of shot than it is for the smaller sizes. For SAE 230 shot, an air pressure of 20 psi is sufficient to give the maximum benefits. Somewhat higher air pressures are needed for SAE 130, and it is likely that somewhat lower air pressure would be required for maximum benefits with SAE 660.

(d) Intensity versus Fatigue Durability – The previous fatigue data are again plotted versus intensity as measured with Almen test strips (see Fig. 11). Here it may be noted that intensity values higher than 0.003C are unnecessary for maximum fatigue durability, and that values lower than 0.003C give decreased benefits. It is not possible from the limited data to distinguish any differences among such minimum values of intensity for the three shot sizes.

(e) Effect of Peening Under Strain – Several groups of specimens were shot peened using various shot sizes under various amounts of surface strain, holding air pressure and exposure constant for each other. During the peening treatment, the specimens were held in a fixture bent to a predetermined curvature while the surface under tensile stress was shot peened. These were subsequently fatigue tested, and the results are shown by Fig. 12. Remarkable increase in fatigue life was obtained – something of the order of 10 to 1 over specimens peened with no strain. Maximum fatigue life seems to be obtained if the specimens are strained during peening to 0.0045 in. per in. or greater for the smaller sizes of shot, and somewhat greater strains for the larger sizes of shot. The yield point corresponds to about 0.0069 in. per in. strain. No explanation was apparent for the fact that a higher surface strain was necessary to obtain maximum fatigue life with the larger sizes of shot.

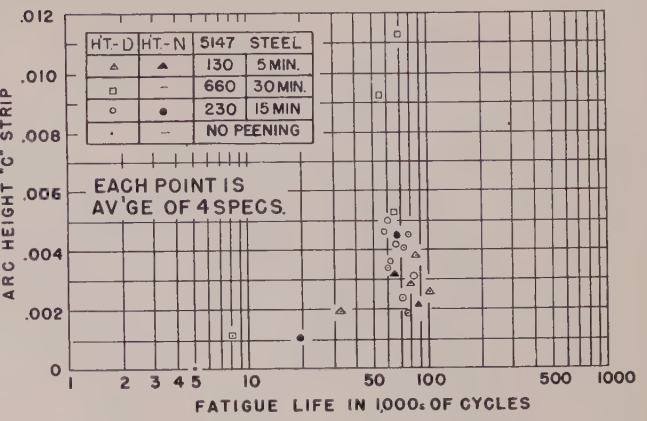


Fig. 11 – Effect of peening intensity on fatigue life

(f) Effect of Low-Temperature Heating After Shot Peening – A group of specimens were shot peened using SAE 230 shot, 50 psi air pressure, 20 min peening time yielding intensities of 0.0037 to 0.0041C. Groups of each were heated to various temperatures up to 400 F for 4 hr. The results are shown by Fig. 13. A gradual reduction in fatigue life with increased temperature up to 400 F seems apparent.

Residual Stress Analysis

1. *Procedure* – (a) Dissection – The stress-analysis specimens were all of the "beam" type parted from the spring specimen by a thin parting wheel on a surface grinder. In the cases where the specimen previously had been fatigue tested and the location of the fatigue origin was known, the stress-analysis specimen was cut from an adjacent location (as shown in Fig. 14). If the spring had not been tested or the origin was not clearly defined, the specimen was taken from the approximate center of the spring. The actual parting operation consisted of four cuts resulting in a specimen 3 in. long, approximately 3/16 in. wide, and the

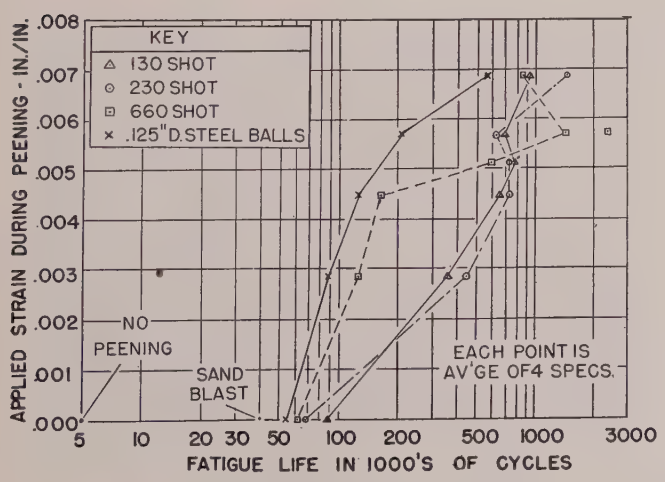


Fig. 12 – Effect of strain peening on fatigue life

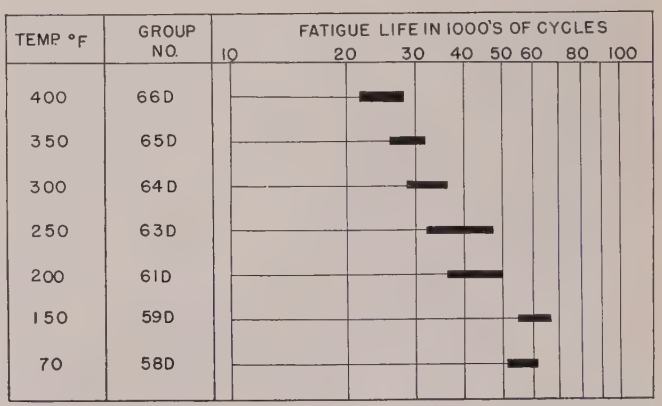


Fig. 13 – Effect of heating specimens for 4 hr. Specimens peened with 230 shot for 20 min at 50 psi and 0.003C intensity

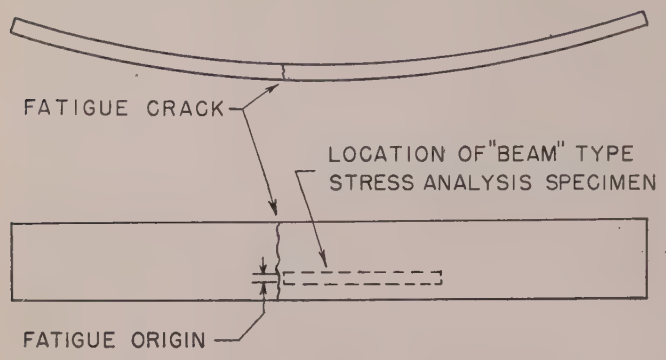


Fig. 14 – Location of residual stress specimen

full thickness of the spring, approximately 0.190 in. (this thickness varied slightly among specimens).

The parted-out piece, to be dissected, was then carefully ground on each side alternately to reduce the width to 0.125 in. This careful grinding was done by an established schedule of very light cuts with frequent dressing of the wheel and with care taken to eliminate overheating of the specimen (as judged by the touch). This process of careful metal removal we shall refer to as "stress grinding" for further discussion.

At this point, the actual dissection to measure the residual stress distribution began. It consisted of carefully removing a series of layers of metal by stress grinding and measuring the change in longitudinal curvature of the specimen for each layer removed.

The actual process of removing a layer was accomplished by placing the specimen in a vise, loading in a press until flat, clamping sides with the vise, and then stress grinding on a surface grinder (Fig. 15). It can be seen that clamping of the specimen in a flat position for each layer removal can only be done if the bending stress introduced into the specimen and the clamping force does not cause any plastic flow or yielding which would change the residual stress distribution.

Before dissection of each specimen was started and after each layer of metal was removed, a very accurate measure of the arc height over a 2.500-in. gage length was made. The arc-height gage that was used for these measurements is shown in Fig. 16. The specimen was placed on knife-edge supports, and contact was made with a dial indicator graduated in 0.0001-in. divisions. However, the use of an electronic microcontactor made it possible to read the arc height to an accuracy of ±0.00002 in. The specimen thickness was also measured with a micrometer and recorded after each layer removal.

The procedure followed on nearly all of the specimens reported included a primary cut (layers stress ground from the unpeened side) of a total of approximately 0.120 in., leaving a metal thickness of 0.070 in. and a secondary cut (layers stress ground from the peened side) with thin (0.001 in.) layers. The number of layers and the depth of each layer removed varied somewhat from one specimen to the next because of the large differences in the depth and magnitude of the stress layer in the various specimens.

Removing the metal first from the unpeened side (primary cut) increased the accuracy of the stress determination of the peened side (secondary cut).

(b) Residual Stress Calculation – The residual stress analysis by this layer-removal method is based on two fundamental conditions that must be satisfied in the unrestrained specimen: namely, internal forces and moments must be balanced. If a layer of metal is removed from a specimen and its curvature changes, there must have been forces present in the layer to hold the specimen in its original shape. Knowing the thickness of the specimen, thickness of the layer, modulus of elasticity, and the curvature change, the force or stress which must have existed in the layer can, of course, be computed.

The scope and subject of this paper does not permit elaboration on the methods used in the actual stress calculation. The formula used is essentially

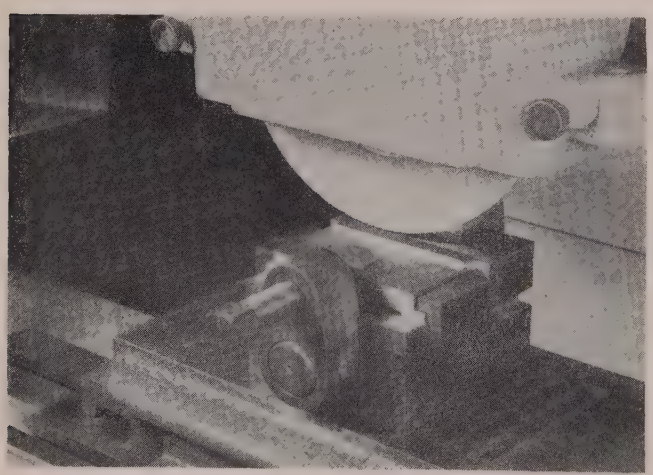


Fig. 15 – Stress specimen in grinding vise

Fig. 16 – Measurement of arc height

the same as that published in the literature.[2, 3] Modifications were made to suit the use of a programmed punchcard digital computer which took most of the labor out of the calculations.

2. *Results of Stress Analysis* – The results of stress analyses of 49 spring specimens are presented in Table 1. A comparison of certain results in order to further understand the effect of heat-treatment, shot peening, and preset on residual stress follows.

(a) As Heat-Treated – Analyses Nos. 36 and 37 are two specimens taken from the same leaf spring (117N-2). They represent the residual stress in the spring specimens as machined and heat-treated. As can be seen in Fig. 17, the stress in the outer layers is compressive with a maximum of 12,000-14,000 psi and extending approximately 0.018 in. deep. It will be shown later that these stresses are relatively small compared to stresses that are induced by shot peening.

(b) Consistency of Results – A logical question arose early in the investigation, specifically, "How reliable are the stress distributions?" In order to answer that question, Fig. 18 is presented, which shows two specimens from each of two groups. These two groups were given different surface treatments, and, consequently, the residual stress distribution is considerably different. Yet within each group, the specimens are quite consistent.

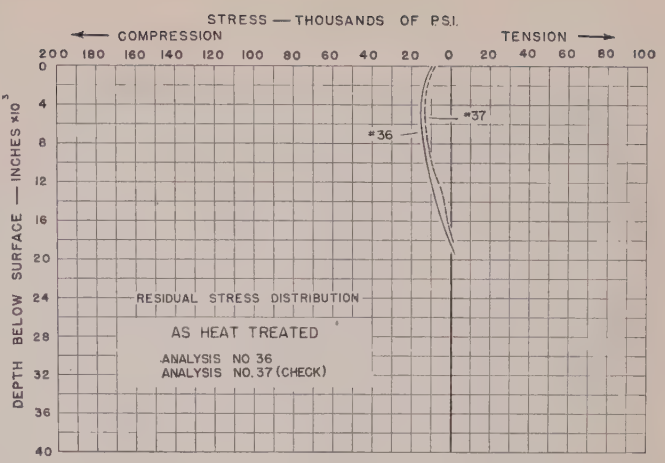

STRESS — THOUSANDS OF PSI.
COMPRESSION
TENSION
200 180 160 140 120 100 80 60 40 20 0 20 40 60 80 100
DEPTH BELOW SURFACE — INCHES ×10³
0 4 8 12 16 20 24 28 32 36 40
#37
#36
RESIDUAL STRESS DISTRIBUTION
AS HEAT TREATED
ANALYSIS NO 36
ANALYSIS NO. 37 (CHECK)

Fig. 17 – Residual stress distribution as heat-treated

(c) Effect of Shot Peening – Four basic shot sizes were used in this investigation. As previously

[2] See Part II of "Strength of Materials," by S. Timoshenko. Pub. by D. Van Nostrand Co., New York, 1941.

[3] See Proceedings of the Society for Experimental Stress Analysis, Vol. III, No. 1, 1945, pp. 40-61: "A Study of Certain Mechanically Induced Residual Stresses," by D. G. Richards.

Table 1 – Tabulated Residual Stress Data

Analysis No.	Spring No.	Steel	Heat-Treat Code No.	Hardness, Rc	Shot Size	Peening Intensity	Peening Surface Strain, in. per in.	Preset Strain, in. per in.	Test Stress, psi	Specimen Life, cycles	Group Average Life, cycles	Max Residual Compression, psi	Depth of Max Stress, in.	Depth of Compression Layer, in.
1	10A-6	5150H	14	48.3–48.8	230	0.016–0.018A	Free[a]	0.0076	197,400	.....	137,434	120,000	0.0023	0.0083
3	44D[a]-1	5147-1	14	47–47.6	660	0.009C	Free[a]	0.0073	.....	.....	.....	109,000	0.0025	0.016
4	44D[a]-2	5147-1	14	47.8–48.2	130	0.0019C	Free[a]	0.0073	.....	.....	.....	112,000	0.0015	0.016
6	42D-4	5147-1	14	45.7–46.3	130	0.0099–0.0101A	Free[a]	0.0073	201,300	83,410	80,760	109,000	0.0015	0.020
7	70D-3	5147-1	14	48.2–48.9	130	0.0013–0.0023C	Free[a]	0.0073	197,400	135,250	144,075	115,000	Surface	0.0150
8	51D-3	5147-1	14	48.1–48.9	230	0.0048–0.0050C	Free[a]	0.0073	198,700	56,500	59,360	110,000	0.002	0.0109
9	59D-2	5147-1	14	48.3–48.6	230	0.0037–0.0039C	Free[a]	0.0073	201,400	57,820	62,740	98,000	0.0012	0.010
10	66D-3	5147-1	14	48.6–49.1	230	0.0034–0.0041C	Free[a]	0.0073	200,000	24,440	25,200	100,000	0.003	0.0102
11	78D-3	5147-1	14	49.6–49.9	230	0.0064–0.0069A	Free[a]	0.0073	199,800	121,650	79,140	96,000	0.0012	0.0113
12	78D-2	5147-1	14	49.5–50.0	230	0.0064–0.0069A	Free[a]	0.0073	200,000	25,530	79,140	99,000	0.001	0.0125
13	79N-2	5147-3	14	47.1–47.4	660	0.0090–0.0091C	0.0054[a]	0.0073	198,200	485,480	2,345,080	139,000	0.004	0.021
14	79N-3	5147-3	14	47.3–47.8	660	0.009–0 0094C	0.0054[a]	0.0073	200,600	3,884,860	2,345,080	+134,000	0.005	0.0202
15	80N-3	5147-3	14	47.9–48.4	660	0.0092C	0.0027C	0.0073	202,700	121,800	128,020	121,000	0.0051	0.0172
16	91N-2	5147-3	14	47.3–47.8	230	0.014–0.017A	0.0027[a]	0.0073	203,800	591,270	547,680	143,000	0.002	0.0105
17	95N-2	5147-3	14	47.4–48.1	130	0.0089A2	0.0054[a]	0.0073	200,000	733,350	697,840	147,000	0.0013	0.0097
18[b]	107-4	4460-1	29	46–46.5	660	0.008–0.012C	0.00633	0.00853	239,000	.....	45,000	142,000	0.006	0.0217
19[b]	109-6	4460-1	31	52–53	660	0.008–0.012C	0.00586	0.00845	236,000	.....	129,000	189,000	0.004	0.019
20	74-1	5150	19	49.5–50.5	660	0.008–0.012C	0.00588–0.00610	0.00840	240,000	.....	589,000	187,000	0.004	0.0233
21[b]	57-6	9262	13	46–47	660	0.008–0.012C	0.00596–Free	0.00868	239,000	.....	32,000	93,000	0.003	0.0226
22[b]	2-6	5150-1	1	45–46.5	660	0.008–0.012C	Free[a]	0.00960	203,000	.....	45,000	92,000	0.0085	0.0215
23[b]	8-5	5150-1	1	46–48	660	0.008–0.012C	0 00516	.....	201,000	.....	157,000	132,000	0.009	0.022
24[b]	17-5	5150-1	1	46.5–47.5	660	0.008–0.012C	Free	0.01662	239,000	.....	14,000	101,000	0.0025	0.030
25	21-4	5150-1	1	45.5–47	660	0.008–0.012C	0.00527	0.01726	243,000	.....	18,000	110,000	Surface and 0.004	0.028
26[b]	16-3	5150-1	1	45–47	660	0.008–0.102C	Free[a]	0.02204	236,000	.....	30,000	105,000	0.005	0.0285
27	100N-4	5147-3	14	48.3–49.2	⅛ balls	0.018–0.020C	0.0068[a]	0.0073	197,000	527,960	560,810	135,000	0.006	0.0323
28[b]	148-5	9262 + 0.20 Mo	68	48.5–49	660	0.008–0.012C	0.00611	0.00923	241,000	.....	63,000	105,000	0.006	0.0205
29	82N-3	5147-3	14	49–49.2	⅛ balls	0.018–0.020C	Free[a]	0.0073	201,600	57,070	59,370	110,000	Surface	0.027
30	87N-3	5147-3	14	48.3	660	0.010C	0.0069[a]	0.0073	200,000	878,000	843,000	148,000	Surface	0.0217
31	10A-5	5150H	14	48.4	230	0.016–0.018A	Free	0.0073	200,000	137,000	137,000	108,000	0.0025	0.0098
32[b]	74-4	5150	19	49.5–50.5	660	0.008–0.012C	0.00588–0.00610	0.00840	240,000	832,000	589,000	165,000	0.0025	0.0215
33[b]	93-6	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	Stress Free	.....	.....
34[b]	86-6	5150	23	44–45	660	0.008–0.012C	0.00585	0.00840	239,000	60,266	60,000	128,000	0.003	0.0212
35[b]	6-1	5150-1	1	45.5–47	660	0.008–0.012C	Free[a]	0.00898	200,000	49,100	48,000	90,000	0.0045	0.021
36 and 37	117N-2	5147-3	14	14	.....	.....	.....	.....	.....	.....	.....	14,000	0.004	0.018
38A and B	117N-1	5147-3	Temper 1200 F	31	.....	.....	.....	.....	.....	.....	.....	38,000	0.003	0.029
39A and B	117N-3	5147-3	14	49	.....	.....	.....	0.0073	.....	.....	.....	18,000	0.0033	0.0185
40A and B	117N-4	5147-3	14	49	.....	.....	.....	0.0083	.....	.....	.....	23,000	0.003	0.017
41A and B	138N-1	5147-3	14	.....	.....	.....	.....	.....	.....	.....	.....	+17,000	0.010	0.027
42	136N-1	5147-3	Requench 14	49.5	.....	.....	.....	0.0092	.....	.....	.....	28,000	0.004	0.021
43	136N-2	5147-3	14	49.5	.....	.....	.....	0.0100	.....	.....	.....	38,000	Surface	0.025
44	136N-3	5147-3	14	49.5	.....	.....	.....	0.0108	.....	.....	.....	39,000	Surface	0.026
45	136N-4	5147-3	14	49.5	.....	.....	.....	0.0117	.....	.....	.....	52,000	Surface	0.030
46	140N-1	5147-3	14	50.0	.....	.....	.....	0.0125	.....	.....	.....	50,000	Surface	0.031
47	140N-2	5147-3	14	49.0	.....	.....	.....	0.0133	.....	.....	.....	42,000	Surface	0.048
48	140N-3	5147-3	14	49.5	.....	.....	.....	0.0142	.....	.....	.....	54,000	Surface	0.033
49	140N-4	5147-3	14	49.5	.....	.....	.....	0.0150	.....	.....	.....	59,000	Surface	0.037

[a] Indicates shot peening followed presetting.
[b] Indicates specimens taken from and investigation conducted by the suspension development group, GM Engineering Staff.

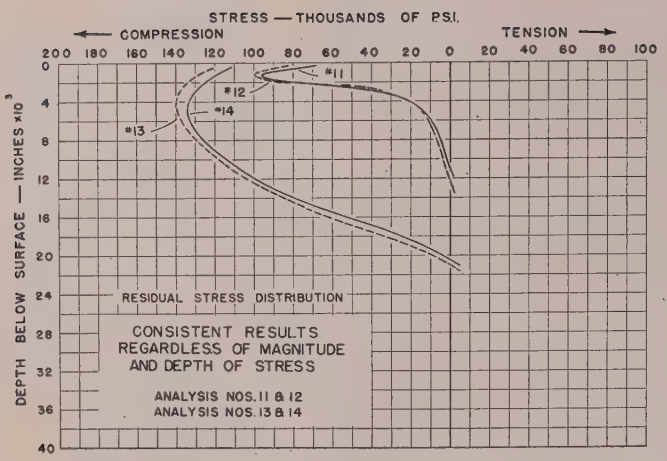
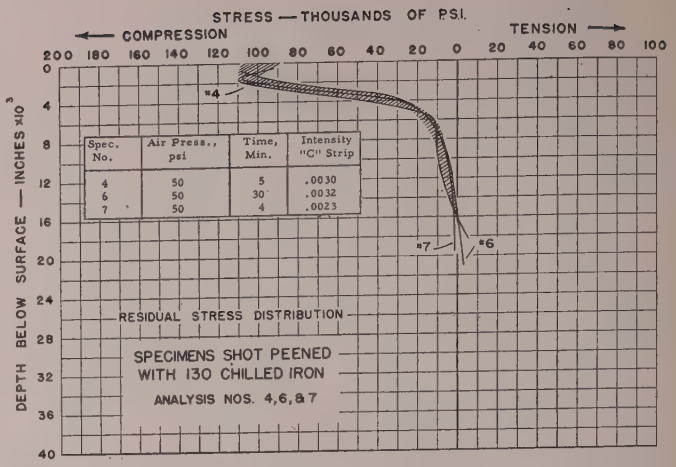


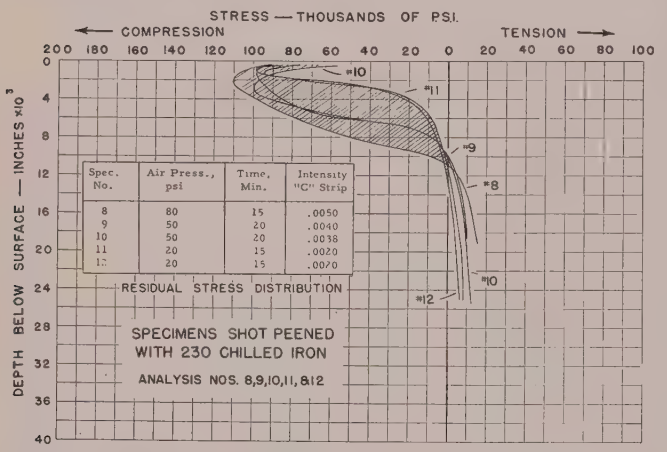
Fig. 18 – Consistency of residual stress distribution results



Spec. No.	Air Press., psi	Time, Min.	Intensity "C" Strip
4	50	5	.0030
6	50	30	.0032
7	50	4	.0023

Fig. 19 – Residual stress in specimens shot peened with 130 chilled iron

discussed, these were 130 (old SAE P-13), 230, and 660 chilled iron, and 0.125-in. diameter steel balls. Fig. 19 shows such a group of specimens that had been peened with 130 chilled iron shot. Individual curves are shown for various air pressures as well as a scatter band within which all specimens fall. Similar bands were constructed for the 230 shot (Fig. 20), the 660 shot (Fig. 21), and a single specimen analysis No. 29 peened with 0.125-in. diameter balls (Fig. 22). In all cases the specimens were given a moderate presetting treatment identical to that used for the fatigue-test investigation. All the specimens were peened to full coverage. It is important to note that the peak compressive stress introduced by peening in all of the specimens was in the range of 90,000 to 115,000 psi. The average of these extremes, 102,500 psi, is approximately 50% of the nominal yield strength. The total depth of the compressive layer, however, varies from approximately 0.006 to 0.026 in., depending on the shot size and velocity (air pressure).

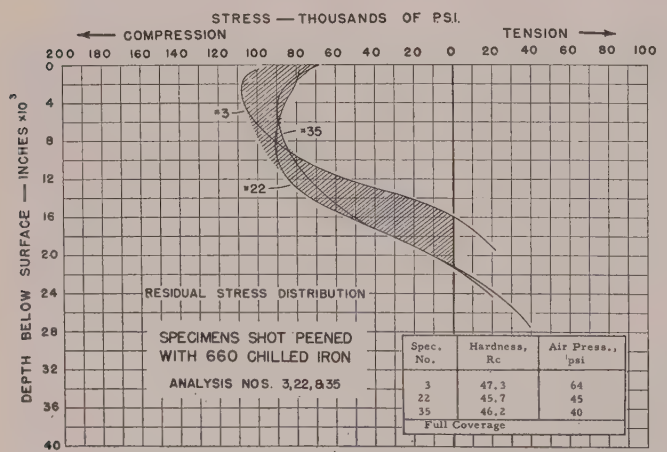


Spec. No.	Air Press., psi	Time, Min.	Intensity "C" Strip
8	80	15	.0050
9	50	20	.0040
10	50	20	.0038
11	20	15	.0020
12	20	15	.0070

Fig. 20 – Residual stress in specimens shot peened with 230 chilled iron

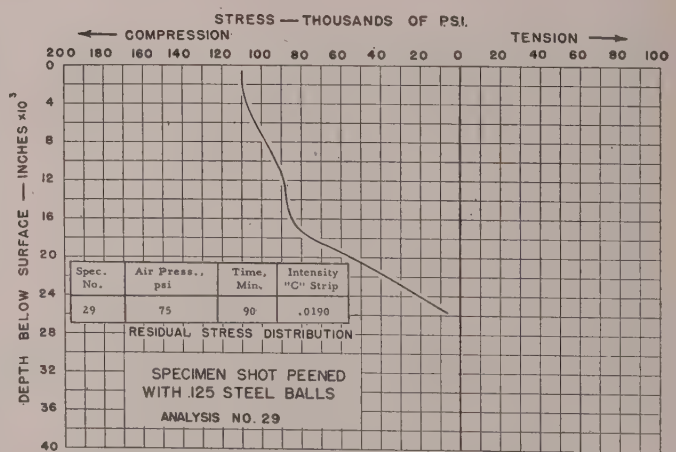
(d) Effect of Presetting with No Peening – In order to understand more fully the effect of presetting on the residual stress distribution in these leaf springs, a series of specimens were prepared and analyzed. This group (Analyses Nos. 39, 40, and 42 through 49) were given the usual heat-treatment, but were preset to varying degrees of surface strain. Fig. 23 shows a few of these specimens. As can be seen, increasing the amount of applied surface tensile strain by bending results in an increasing amount of residual compressive stress.

(e) Strain Peening – The addition of a tensile



Spec. No.	Hardness, Rc	Air Press., psi
3	47.3	64
22	45.7	45
35	46.2	40
Full Coverage		

Fig. 21 – Residual stress in specimens shot peened with 660 chilled iron



Spec. No.	Air Press., psi	Time, Min.	Intensity "C" Strip
29	75	90	.0190

Fig. 22 – Residual stress in a specimen shot peened with 0.125-in. diameter steel balls

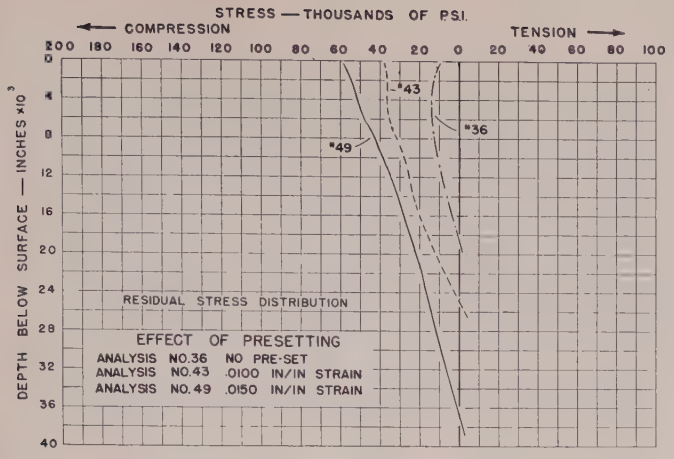


Fig. 23 – Effect of presetting on residual stress

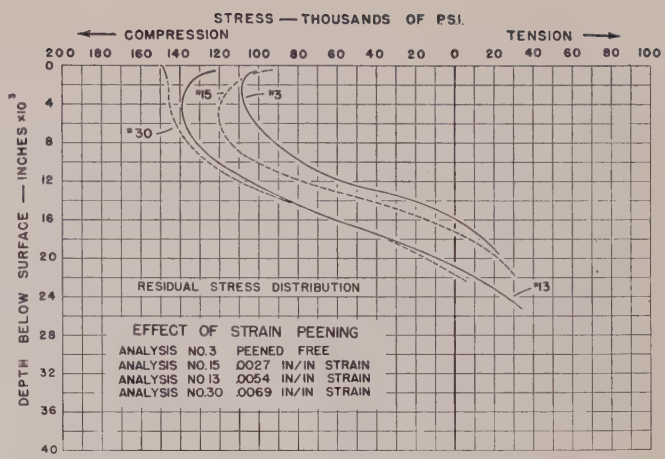


Fig. 24 – Effect of strain peening on residual stress

strain to the surface during shot peening has the effect of increasing the magnitude and depth of the resulting residual compressively stressed layer. See Fig. 24. The magnitude of residual stress is increased by as much as one third by peening under strain.

The depth at which the maximum stress occurs apparently increases with increase in surface strain during peening.

(f) Cold Working versus Hardness – To learn more about the effect of specimen hardness on the resultant residual stress pattern caused by shot peening, two 4460 steel specimens were selected and analyzed (Analyses Nos. 18 and 19). The specimens had been given different draw temperatures, resulting in hardnesses of Rockwell C 46 and 52 respectively. The specimens were given identical peening treatments (660 chilled iron while under strain) and preset treatments. The residual stress results are shown in Fig. 25. The harder specimen contained the higher residual stress due to the higher yield point. Also, the depth of the stressed layer was found to be less than that found in the softer specimens.

At this point, for comparative purposes, it seems worth while to show the wide range of magnitude and depth of the residual stresses that can be obtained by cold working steel with various peening methods. Fig. 26 shows a shot-peened ball bearing (52100 steel, approximately Rockwell C 62), a spring specimen (Analysis No. 35, approximately Rockwell C 46), and an annealed 1020 specimen (air-hammer peened, approximately Rockwell B 75). The effect of hardness is apparent both on the magnitude and the depth of the residual stress.

(g) Effect of Fatigue Testing – Analyses were made on both fatigue-tested and untested specimens from the same group. These were made to determine the effect of fatigue testing on the residual stresses in the spring specimens. A typical comparison is shown in Fig. 27. It appears that some yielding occurs during fatigue testing which results in a reduction of the compressive stress.

Correlation Between Residual Stresses and Fatigue Life

In conclusion, we wish to correlate the data from the two phases of the investigation. A look at Fig. 28 indicates that when bending stresses are added to the residual stress due to shot peening, the resultant stresses in the surface layers are considerably reduced. That is, in fatigue loading, these

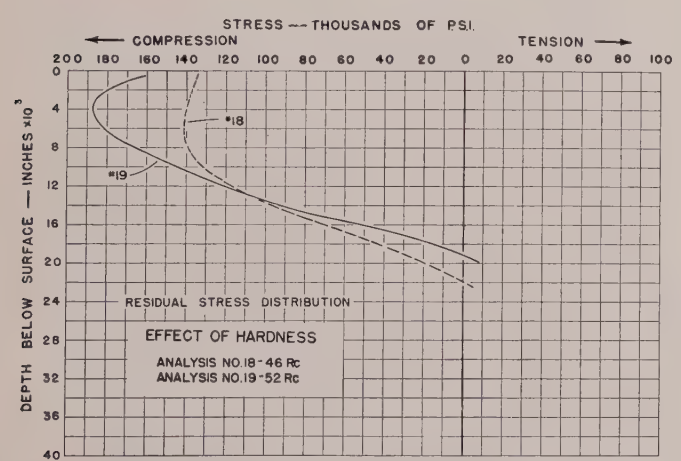


Fig. 25 – Effect of hardness on residual stress

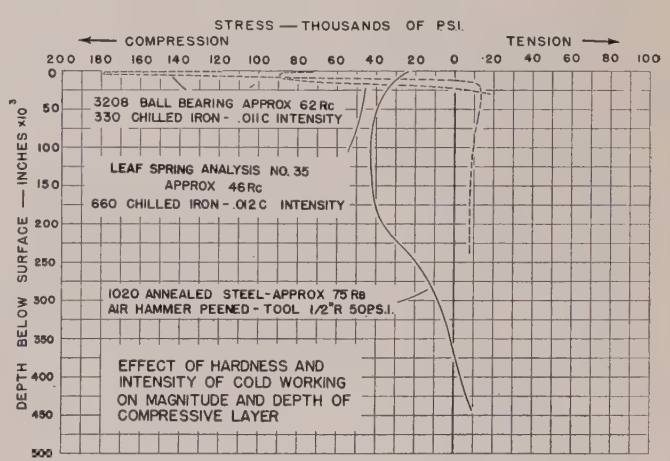

STRESS — THOUSANDS OF P.S.I.
COMPRESSION
TENSION
DEPTH BELOW SURFACE — INCHES ×10³
3208 BALL BEARING APPROX 62 Rc
330 CHILLED IRON - .011C INTENSITY
LEAF SPRING ANALYSIS NO. 35
APPROX 46 Rc
660 CHILLED IRON - .012 C INTENSITY
1020 ANNEALED STEEL - APPROX 75 Rb
AIR HAMMER PEENED - TOOL 1/2"R 50PS.I.
EFFECT OF HARDNESS AND
INTENSITY OF COLD WORKING
ON MAGNITUDE AND DEPTH OF
COMPRESSIVE LAYER

Fig. 26 – Effect of hardness and intensity of cold working on magnitude and depth of compressive layer

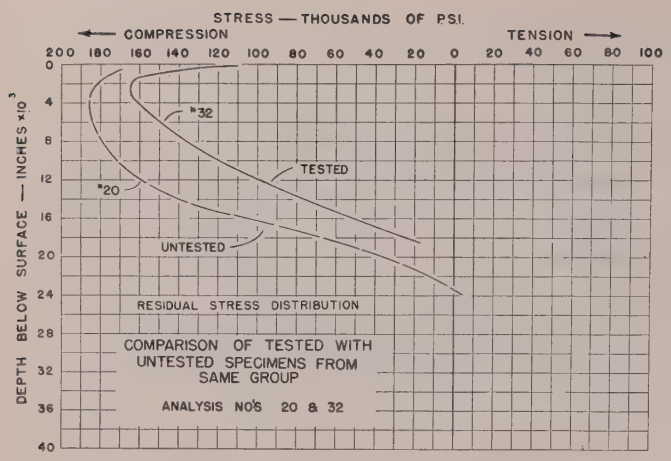


Fig. 27 – Effect of fatigue testing on residual stress

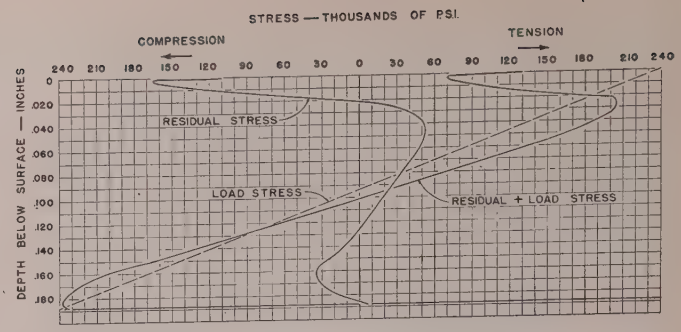


Fig. 28 – Combination of residual stress and bending stress; residual stress analysis 32. Stress through complete thickness of specimen is shown

surface layers are experiencing a much lower peak stress than would be the case had not shot peening been used. Since a surface may be considered to be a stress raiser due to roughness, cracks, and the like, the protection at the surface is very beneficial as indicated by the fatigue data.

The relationship between fatigue properties and residual stresses can be shown by plotting "net" stress versus fatigue life. "Net" stress may be defined as the algebraic sum of the residual stress and load stress. Taking the previous data and plotting accordingly, we have Fig. 29. For this plot, residual stress was taken at its peak compressive value, and load stress was taken as the stress at the corresponding distance below the surface due to the applied load.

It will be noted that the data take the shape of the familiar S-N curve. This serves to show that high residual compressive stresses are desirable for this type of loading.

Although "work hardening" may have played some part in the improvement obtained in fatigue life by the various cold-working treatments, these results indicate residual stress plays an important and definite part.

On this basis, it is quite understandable why the fatigue life varied from 5000 cycles for the non-peened specimens to nearly a million cycles for the shot-peened specimens, despite the fact that the test stress for all specimens was 200,000 psi.

In an effort to explain the scatter obtained in fatigue tests, two specimens from one group were analyzed which had drastically different fatigue lives. The results are shown by Fig. 30. Since there is no apparent difference between the two, the indication is that the residual stresses as measured had little to do with the wide difference in fatigue life obtained.

Acknowledgment

The research work outlined in this paper was conducted by the engineering mechanics department of the General Motors Corp., Research Laboratories Division, as a part of its regular work in the fields of fatigue, cold working of metals, and residual stresses. Valuable assistance was obtained from the structure and suspension development department of the Central Engineering Staff, and the metallurgy, processing, and special problems departments of the Research Laboratories Division. To all personnel involved, the authors wish to express their appreciation and thanks.

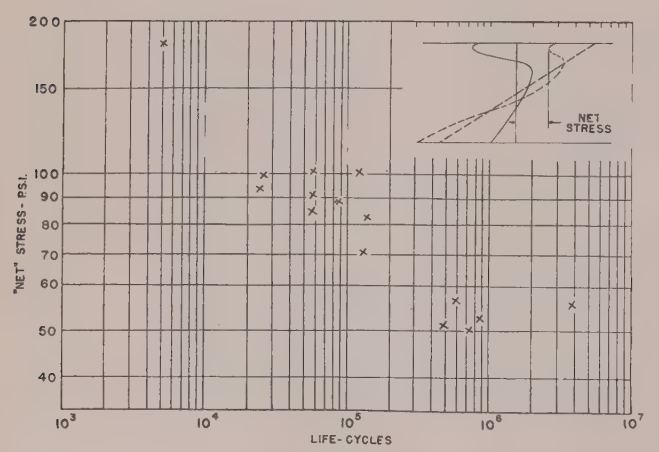


Fig. 29 – Plot of test stress, maximum residual stress versus life

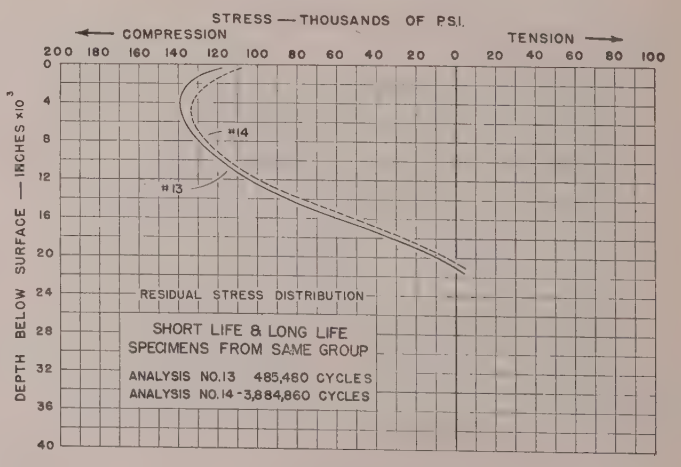


Fig. 30 – Residual stress in long-life and short-life specimens from same group

Effect of Temperature on Endurance Limit and Relaxation of Spring Materials

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W. P. Wood, University of Michigan

This paper was presented before Division XX of the SAE Iron and Steel Technical Committee, Hershey, Pa., Oct. 1, 1953.

MECHANICAL springs, with increasing frequency, are required to operate at temperatures other than those in the usual atmospheric ranges. A knowledge of the changes in physical and mechanical properties is necessary under these circumstances in order to arrive at proper design and specifications for the springs. It is well known that the torsional modulus and relaxation under static loading vary with temperature, but practically nothing has been known of the fatigue characteristics of spring materials at low temperatures and in the range from 75 F to 700 F. Further interest in this connection arises from the question as to what degree the beneficial effects of shot peening on fatigue life of springs will persist at somewhat elevated temperatures.

The investigation herein reported was therefore undertaken to ascertain (1) the maximum stresses which can be sustained and (2) the losses in load-carrying capacity, or set, of various spring materials, at temperatures above and below atmospheric. The lowest temperature at which any tests were made was —75 F and the highest was 650 F. The majority of the tests were made in the range from 70 F to 450 F. Endurance tests were made on all materials in both the shotpeened and the unpeened condition. Relaxation tests were made only on unpeened springs since shot peening produces

ENDURANCE limits and load losses at various temperatures from –75 F to 650 F for several spring materials are reported in this paper. The materials tested were in the form of helical springs, both shot peened and unpeened.

Some general observations made by the authors include:

1. Shot-peened springs had higher endurance limits and greater relaxation than unpeened springs at –75 F and 75 F.

2. As test temperatures rose above atmospheric, endurance limit of all unpeened springs tended to hold steady or increase somewhat, while that of shot-peened springs tended to decrease.

3. Except for high-speed steel and stainless steel, chrome-silicon steel showed lowest load loss in both static and dynamic tests in the temperature range from atmospheric to 450 F.

4. Both unpeened and shot-peened high-speed-steel springs showed high endurance limits at all test temperatures.

5. Endurance limits at –75 F were similar to those at atmospheric temperature, but the amount of set was less, on the average.

The Authors

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W. P. WOOD is professor of Metallurgical Engineering and Supervisor of Research for the Associated Spring Corp. at the University of Michigan. He received B.S. and M.S.E degrees in Metallurgical Engineering from the University of Michigan.

trapped surface stresses and the springs lose some stability when heated, resulting in an apparent lower resistance to set. All tests were made on steels in the form of helical springs.

Prior to beginning the investigation, a search of the literature failed to reveal any references regarding the effect of temperature upon the fatigue life of springs; but later it was found that in 1927 Lea and Heywood[1] had published the results of an extended study of the failure of steel wire under repeated torsional stress, the wire being in the form of helical springs. While most of their tests were carried out at atmospheric temperatures, a few endurance runs on single springs were made at 100 C (212 F) and 150 C (302 F). They reported that an 0.84% carbon-steel spring showed a somewhat lower safe stress at 100 C than at room temperature, while a chrome-vanadium-steel spring showed a higher safe stress at 150 C than at room temperature. No actual values were given for the latter test, and no details of temperature control or variation in the heater were supplied. The tentative conclusion stated that safe stresses at 100 C and 150 C did not appear to differ materially from those at ordinary temperatures.

In 1950, a few years after the present studies were started, Pomp and Hempel[2, 3] published the results of endurance tests of helical steel springs at 250 C (482 F) as compared to results at room temperature. They investigated four types of spring steel: (1) a 0.67% plain carbon steel, (2) a chrome-vanadium steel, (3) a chrome-silicon steel, and (4) a silico-manganese steel. Part of the tests were on springs made from pretempered wire, the remaining tests being carried out on springs heat-treated after coiling, with the supposition that annealed wire was used in coiling the springs. In both these groups tests were made on shot-peened and unpeened springs.

While, in general, Pomp and Hempel's work indicated the same trends as herein reported, there were such marked variations in their results as to lead one to examine closely their procedure regarding pretreatment of the springs, and so forth. The Rockwell C hardness range of the pretempered wire was 45 to 47 in the four steels used. In the case of the springs heat-treated after coiling, the hardness range was 47 to 57 in the same three alloy steels (no tests were made on heat-treated carbon-steel springs). This higher hardness range was undoubtedly due to the much lower drawing temperature which was used, as compared to that used in preparing the pretempered wire. This probably accounts for the lower endurance limits of the heat-treated springs at room temperature, as compared to springs made from pretempered wire. There was one exception, namely, the chrome-silicon steel, comment on which will be made later.

Shot peening was done with "glass-hard" steel pellets 0.5 mm in diameter, at a blast pressure of about 88 psi and a blasting period of 30-45 min. After shot peening, the springs were reheated for 30 min at 590 F for removal of residual stresses and to raise the elastic limit. It would be presumed that the unpeened springs of pretempered wire were given a bluing heat after coiling, but nothing was said about it.

As a final preliminary step, all shot-peened springs and unpeened springs made from pretempered wire were given a lacquer coating of a Japan type, which involved heating them for 30 min at 350 F. This was not done in the case of the unpeened heat-treated springs since they had a black oxide coating from the final draw heat. Prevention of corrosion as a factor in the test would account for this rustproofing operation.

Pomp and Hempel imply that conditions during the shot-peening operation may not have been of the best and cite the results obtained on the chrome-silicon springs made from pretempered wire as an example. At room temperature, the shot-peened springs showed an increase in fatigue life of only 7%, while the carbon steel had an increase of 22% and the silico-manganese an increase of 86%. Such irregular results, it would seem, could only be explained by variable conditions in the shot-peening operation. Certainly the quality of the steels would not vary that much, since all the wire was made from carefully processed billets. It would seem that the distance of the nozzle from the springs during peening could well be held responsible for these differences, as the intensity would vary with the square of the distance.

Five factors militate against detailed comparison of the results of Pomp and Hempel's work with those of the present investigation. These factors are: (1) nonuniform conditions in their shot-peening operation, (2) higher hardness values of their heat-treated springs as compared to springs made from pretempered wire, (3) the use by the present authors of a relaxation or set limit (15%) as an end point, while Pomp and Hempel used spring failure, (4) greater temperature variation of the springs during test, and (5) no modulus corrections were made. The trends observed in this German work, however, furnish very definite checks on the results reported herein. For example, in five cases out of seven the unpeened springs exhibit higher endurance limits at 482 F than at room temperature. The two exceptions are the chrome-silicon and the carbon-steel springs made from pretempered wire. The chrome-silicon need not be considered for the results were admittedly

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[1] See Proceedings of the Institution of Mechanical Engineers, Vol. 1, 1927, pp. 403-463: "Failure of Some Steel Wires under Repeated Torsional Stresses at Various Mean Stresses Determined from Experiments on Helical Springs," by F. C. Lea and F. Heywood.

[2] See *Archiv für das Eisenhüttenwesen,* Vol. 21, July-August, 1950, pp. 263-272: "Dauerfestigkeit von Schraubenfedern bei erhöhter Temperatur," by A. Pomp and M. Hempel.

[3] See *Archiv für das Eisenhüttenwesen,* Vol. 21 July-August, 1950, pp. 243-262: "Dauerfestigkeit von Schraubenfedern unterschiedlicher Fertigungsart," by A. Pomp and M. Hempel.

unreliable. The carbon-steel springs show lower endurance limits at 482 F than at room temperature. In the present investigation, while the carbon-steel springs showed higher endurance limits at 400 F than at room temperature, the values began to drop off sharply about 400 F, having notably decreased at 450 F, above which temperature no tests were made. Secondly, at 482 F the endurance limit of the shot-peened springs made from pretempered wire had dropped down practically to the values of the unpeened springs at that temperature. The shot-peened alloy-steel springs heat-treated after coiling showed some increase in endurance limit at 482 F, but the room-temperature values were considerably lower than for springs made of pretempered wire, so it is difficult to reconcile these results. Then, too, the hardness of these springs was several points higher than that of the pretempered wire, and this may have been a factor in the varying results.

Pomp and Hempel did not offer any explanation for the increases in endurance limit other than to suggest that the higher hardnesses of the alloy steels might account for them.

Further evidence regarding the effect of temperature upon endurance limit is furnished by Moore and Jasper.[4] They reported that a normalized 0.49 carbon steel and a hardened chrome-nickel steel both showed an increase in endurance limit as the temperature rose to 900 F in the case of the carbon steel and to 500 F in the case of the chrome-nickel steel. These results were obtained by bending tests. They attributed these results to a "healing" effect, which could probably also be termed stress relief. From their tests, however, they thought no general conclusion could be drawn.

[4] See p. 9 of University of Illinois Engineering Experimental Station, Bulletin No. 152, 1925, "Investigation of Fatigue of Metals – Series of 1925" by H. F. Moore and T. M. Jasper.

Materials Investigated

Springs tested were made from the following materials:

1. Pretempered carbon-steel wire.
2. Pretempered chrome-vanadium-steel wire.
3. Pretempered chrome-silicon-steel wire.
4. Pretempered high-manganese-steel wire.
5. Hard-drawn 18 chrome – 8 nickel stainless-steel wire.
6. Patented music wire.
7. High-speed-steel wire, springs being heat-treated after coiling.

The high-speed-steel wire was originally received as straight lengths, and springs were wound on a lathe. It was then found after hardening and tempering that the hardness was Rockwell C 47, which was at least 5 points low. The coiled springs were accordingly given the following heat-treatment:

1. Pack annealed at 1575 F and cooled at 25 F per hour to 900 F.
2. Hardening preheat at 1500 F.
3. Hardening high heat at 2250 F in a Hayes furnace.
4. Quenched in oil.
5. First draw 1050 F for 3 hr.
6. Second draw 1150 F for 3 hr.

Table 1 presents all the data concerning composition, mechanical properties, and preliminary treatments of the springs.

Procedure

Fatigue Tests – All fatigue tests were made on specially constructed machines which subjected the springs to a series of calculated deflections.

Table 1 – Chemical Composition, Mechanical Properties, and Preliminary Treatment of Spring Steels

Type of Steel	Dia of Wire, In.	Tensile Strength, psi	Rockwell C Hardness	Carbon, %	Silicon, %	Manganese, %	Phosphorous, %	Sulfur, %	Chromium, %	Vanadium, %	Nickel, %	Tungsten, %	Preliminary Treatment	
													Shot Peened	Not Shot Peened
Carbon-steel valve tempered	0.148	222,800	44–45	0.67	....	0.84	0.012	0.028	....	....	....	....	750 F for ½ hr shot peened ½ hr 450 F for ½ hr	750 F for ½ hr
Tempered chrome-vanadium	0.148	240,000	48–49	0.50	....	0.72	....	....	0.97	0.17	....	....	750 F for ½ hr shot peened ½ hr with cut-wire shot 450 F for ½ hr	750 F for ½ hr (wire dia 0.142 in.)
Tempered chrome-silicon	0.135	245,249	49	0.60	1.48	0.78	0.012	0.021	0.60	....	....	....	750 F for ½ hr shot peened ½ hr 450 F for ½ hr	750 F for 1 hr
High-manganese valve	0.148	245,000	48	0.72	....	1.44	0.015	0.021	....	....	....	....	750 F for ½ hr shot peened ½ hr 450 F for ½ hr	750 F for ½ hr
18-8 stainless	0.148	213,218	42	0.02	....	0.40–0.45	0.005–0.007	0.014–0.018	18.2–18.8	....	8.85–8.89	....	850 F for ½ hr shot peened ½ hr 450 F for ½ hr	850 F for ½ hr
High-speed	0.148	.....	52–53	0.70	0.35	0.25	0.004	0.012	3.76	1.20	0.07	16.09	Shot peened ½ hr 450 F for ½ hr	No treatment since heat-treated after coiling
Music wire	0.148	259,000	45[a]	0.91	....	0.43	0.018	0.023	....	....	....	....	725 F for ½ hr shot peened ½ hr 450 F for ½ hr	750 F for ½ hr

[a] This hardness value was obtained on finished springs.

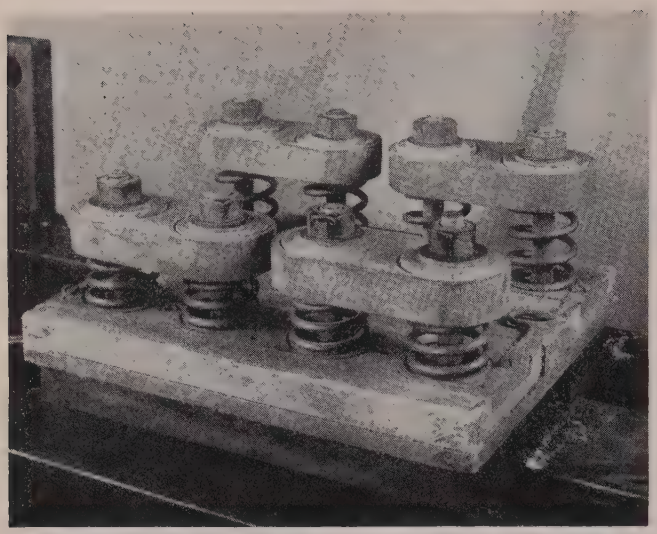

Fig. 1 – Mounting of springs and insulating base plate

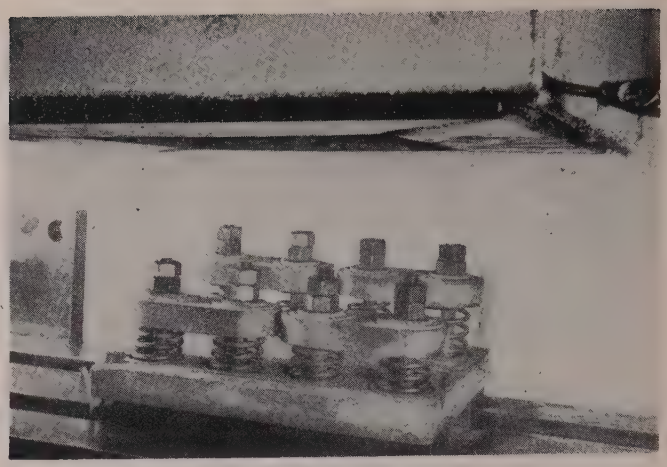

Fig. 2 – Furnace suspended above springs, ready to be lowered around base plate

Rectangular box-type furnaces were constructed for each machine. The furnaces were suspended over each machine in such a manner that they could be lowered over the springs while the machines were running, thus making it possible to operate the test at any desired temperature. Exploratory tests were made during early operations, and it was established that the temperature variation in the operating chambers was less than ±5 F. Cenco bimetallic-type thermostatic controllers were used to regulate the temperature.

Fig. 1 is a close-up showing the mounting of the springs and the insulating base plate around which the furnace fitted during a run. Fig. 2 is another close-up showing the furnace suspended above the springs and ready to be lowered around the base plate. Fig. 3 is a general view showing the furnace suspended above the machine. In Fig. 4 the furnace is in position for making a run. In all cases runs were not started until the springs had been heated to the desired temperature.

Each machine was fitted with an automatic relay to stop the run on failure of any one spring. Relays operated by oil pressure protected each machine from damage by failure of the lubricating system. Each machine subjected the springs to 1740 compressions per minute.

The standard length of each test was set as 10,000,000 cycles or compressions. This corresponds to about 96 hr of running time. Eight springs were tested simultaneously at the same stress, and if one failed all eight were stopped at that point. In many tests spring failure did not occur, and in that case, springs were considered to have run successfully if they did not sustain more than a 15% loss in load. In other words, a load loss of 15% or more was considered as a failure, and the stress was lowered 5000 psi for the next test at the same temperature.

The Wahl stress formula, with curvature and shear corrections, was used to compute the total stress in each spring. The equation was solved for the load in pounds to produce a given stress:

$$\text{Load} = \frac{Sd^3\pi}{8DK}$$

where:

S = Stress, psi, corrected

d = Diameter of wire, in.

D = Mean diameter of coil, in. (outside diameter minus the wire diameter)

K = Wahl correction factor $\left(\frac{4c-1}{4c-4}\right)+\left(\frac{0.615}{c}\right)$,

where $c = \frac{D}{d}$

The computed load was then applied to each spring by spring-testing scales, and the initial and final deflections obtained by an Ames dial micrometer. The initial stress was kept constant at 40,000 psi for all runs. This stress was necessary at the higher running temperatures to maintain electrical contact between the springs and the steel caps so that the relay control current would not be broken except by spring failure. The springs were then assembled in the machines with the same deflections as those determined by the dial micrometer.

Load losses were determined after 1 hr and 24 hr of running time and at the end of the run, provided failure had not occurred. These losses were determined in the following manner: After removal from the machine, the springs were subjected to loading on the same scales as previously used and the load in pounds obtained necessary to cause the same deflection as the load when the springs were originally installed in the machine. The difference between this load and the load causing the same deflection originally constituted the load loss and was converted into per cent loss based on the original load. It was observed that the springs sustained most of the load loss during the

first 24 hr of running. After the load loss was obtained, the springs were reassembled in the machine at the same height and with the same stroke as when originally installed, thus making conditions the same as if they had run continuously through the whole test period and had not been removed at intervals. This procedure also made the various load losses cumulative, that is, the total load loss was the sum of the load losses ascertained during the run.

If a spring broke at any time during the run, the stress was reduced by 5000 psi and a new run started. Similarly, if a total load loss of 15% or over was sustained by the end of the run or earlier, it was rated as equivalent to a spring failure.

For the low-temperature endurance tests, the procedure was the same except that the furnace on one of the machines was replaced by a chamber whose temperature was maintained at −75 F. This chamber was constructed with two compartments, an inner one to hold dry ice and an outer one, 3 in. in thickness, which was filled with granulated cork for insulation. The inner wall of the dry-ice compartment consisted of wire screen and the outer wall of ½-in. fiberboard. With this construction no difficulty was encountered in maintaining the temperature of −75 F for periods of 12 to 14 hr. The temperature of the dry-ice chamber was measured by means of a copper-constantan thermocouple, calibrated for temperatures below the ice point.

Relaxation Tests – These tests were static load-loss tests in which springs were compressed to a calculated stress by means of bolts threaded at each end and fitted with collars. Only springs which were not shot peened were used in these tests. They were then placed in ovens and held at temperature for 72 hr, after which load losses were calculated as previously explained. Five springs were used in each test. This procedure was described in detail by F. P. Zimmerli.[5]

Modulus Corrections – Since the modulus in torsion of steel decreases as temperature increases, spring stresses calculated at room temperature would be higher than the actual operating stresses during the fatigue runs at elevated temperatures. Since a direct proportionality exists between spring loads, spring stresses, and the modulus, corrections for the decrease in modulus were made in the following manner: A spring was deflected on the scales to a convenient value at room temperature

[5] See ASME Transactions, Vol. 63, May, 1941, pp. 363-368: "Effect of Temperature on Coiled Steel Springs under Various Loadings," by F. P. Zimmerli.

Fig. 3 – General view of furnace suspended above machine

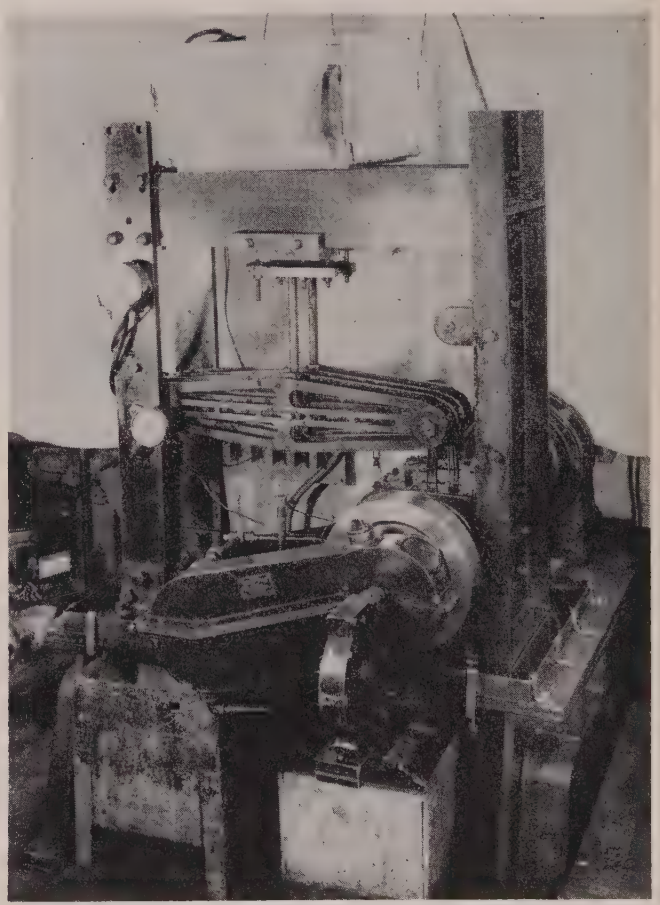

Fig. 4 – Furnace in position for making a run

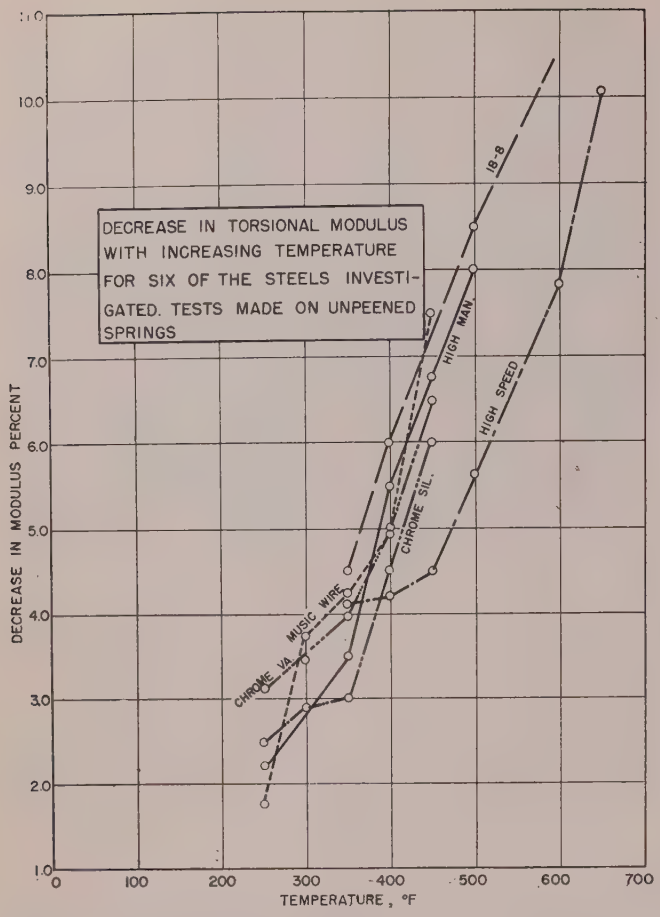

Fig. 5 – Decrease in torsional modulus with increasing temperature

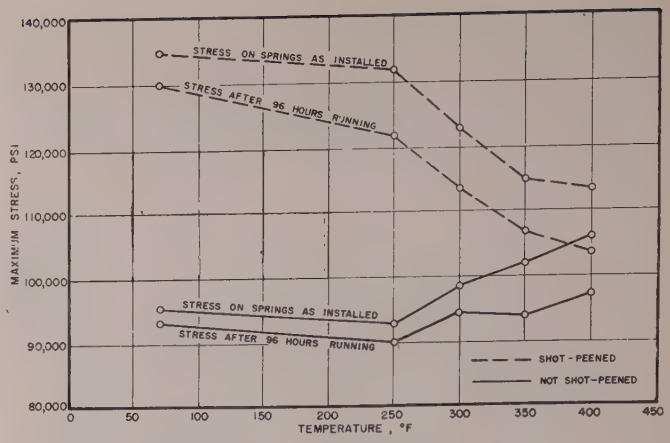

Fig. 6 – Operating stresses and temperatures permissible in carbon valve spring steel for infinite load applications

and the load recorded. A tubular heater was then placed around the spring, and the spring was heated until the desired temperature was reached. The spring was then loaded until the same deflection as before was reached. This load is, of course, less than that required at room temperature. This load under temperature, divided by the load at room temperature, gave a factor by which the room-temperature stress could be multiplied in order to give the stress at the actual running temperature of the test.

Fig. 5 is a graph showing the per cent decrease in torsional modulus with increasing temperature of six of the steels used in these tests. The carbon steel is not included, but its values fall in the midst of the same family of curves. It may be noted that, with the exception of high-speed steel, there is nowhere a variation of over 1.5% in the modulus decrease at any given temperature.

Results and Discussion: Springs at Elevated Temperatures

The results of this investigation are presented in a series of 14 graphs (Figs. 6-19 inclusive), two for each steel. The first graph in each pair shows the variation in endurance limit with temperature of the shot-peened and unpeened springs. In each case the upper curve indicates stress on springs as installed with modulus correction where necessary, while the lower curve represents the load loss in terms of stress after 10,000,000 compressions.

The second graph in each pair is a plot of static load losses with the endurance limit of the unpeened springs plotted on the same graph. This combination of data gives a useful picture of the

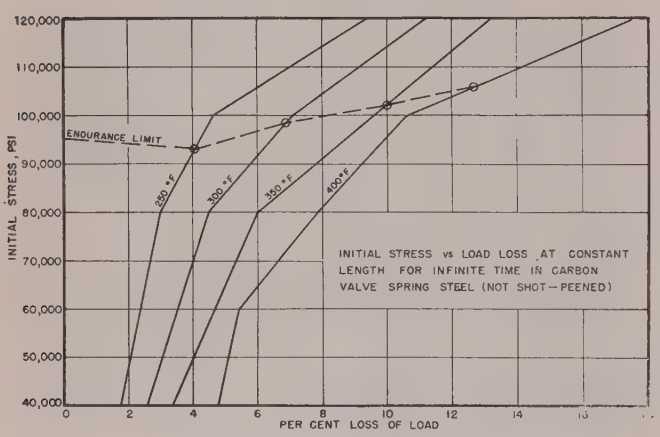
ENDURANCE LIMIT
250 °F
300 °F
350 °F
400°F
INITIAL STRESS vs LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME IN CARBON VALVE SPRING STEEL (NOT SHOT-PEENED)
INITIAL STRESS, PSI
PER CENT LOSS OF LOAD

Fig. 7 – Fatigue and heat set data

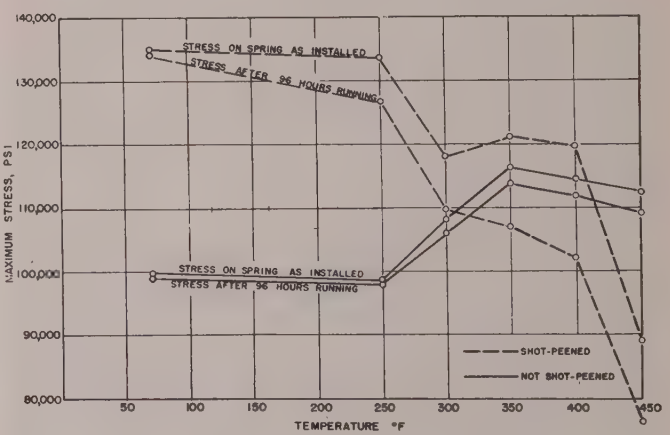

Fig. 8 – Operating stresses and temperatures permissible in chrome-silicon valve spring steel for infinite load applications

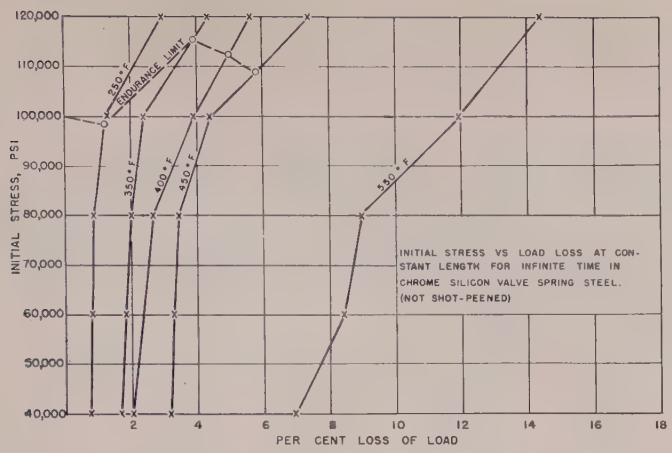
INITIAL STRESS, PSI
PER CENT LOSS OF LOAD
ENDURANCE LIMIT
350° F
400° F
450° F
550° F
INITIAL STRESS VS LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME IN CHROME SILICON VALVE SPRING STEEL. (NOT SHOT-PEENED)

Fig. 9 – Fatigue and heat set data

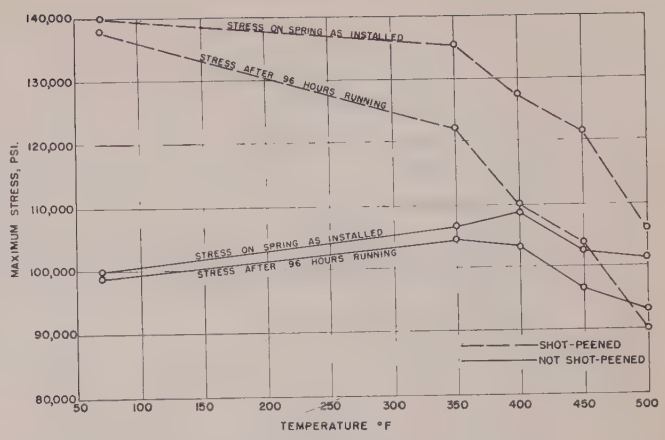

Fig. 10 – Operating stresses and temperatures permissible in high-manganese valve spring steel for infinite load applications

utility of springs under heat and varying loads.

As temperatures increase above atmospheric, the effect of shot peening becomes progressively less, and the endurance limits of these springs approaches those of the corresponding unpeened springs, which latter values have shown little tendency to decrease; in the case of the pretempered wire, all have higher endurance limits at 400 F than at atmospheric temperature. Also evident are the greater load losses which have been sustained by the shot-peened springs.

The endurance limits of the unpeened steels, which were initially in a highly stressed condition, for example, music wire and 18-8 stainless steel, show a slight downward trend but at a lower rate than is the case of the same steels in the shot-peened conditions. At 350 F the 18-8 unpeened springs show a slightly higher endurance limit than at room temperature.

Of the pretempered steels, the chrome-silicon shows the lowest load losses on the average.

The 18-8 stainless steel did not sustain as high stresses as the other alloy steels but, with the exception of high-speed steel, held up better in the range from 400 to 600 F.

It was expected that the high-speed-steel springs would show some superiority to the others at elevated temperatures, but the results were well beyond expectations. Tests were carried out at a maximum temperature of 650 F, since that was as high as the furnace would go, but even there the effect of the shot peening was still present, and the endurance limit was 130,000 psi for the shot-peened springs. The unpeened springs showed only a small drop in stress sustained at 650 F and gave some evidence that if a higher temperature could have been reached the endurance limit would have increased, as was the case at lower temperatures for the springs made from pretempered wire.

Some of the points on the endurance curves appear to be out of line, notably in the case of shot-peened springs of chrome-vanadium steel and music wire at 350 F. Such departures from the general direction of the curves may be errors introduced by general test operations, such as wrong height adjustment with calipers, bending in the crossbars of the machine, or stretching of the connecting rods. Then, too, the caps holding the springs may have worn excessively before being replaced, although this latter condition was checked

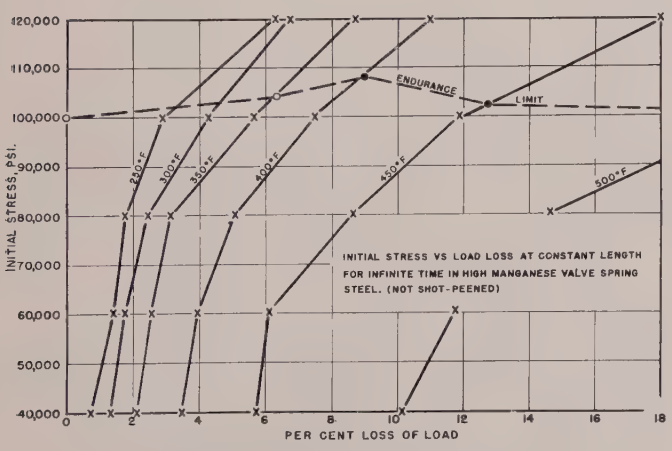
INITIAL STRESS, PSI.
PER CENT LOSS OF LOAD
ENDURANCE LIMIT
250° F
300° F
350° F
400° F
450° F
500° F
INITIAL STRESS VS LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME IN HIGH MANGANESE VALVE SPRING STEEL. (NOT SHOT-PEENED)

Fig. 11 – Fatigue and heat set data

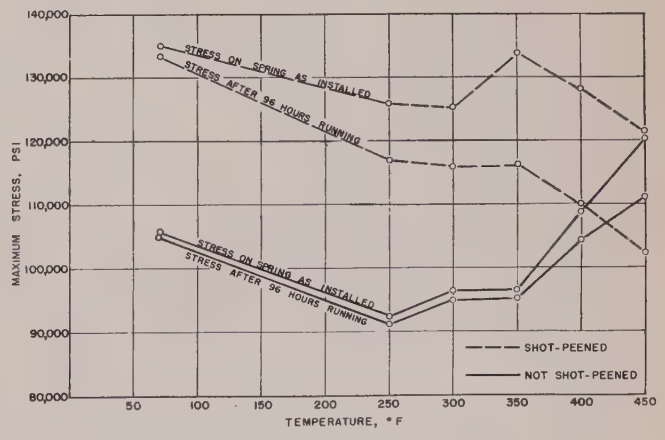

Fig. 12 – Operating stresses and temperatures permissible in chrome-vanadium valve spring steel for infinite load applications

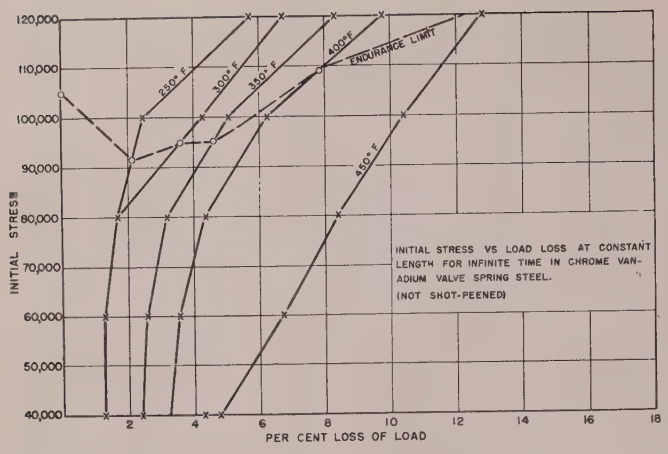
INITIAL STRESS
PER CENT LOSS OF LOAD
250° F
300° F
350° F
400° F
450° F
ENDURANCE LIMIT
INITIAL STRESS VS LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME IN CHROME VANADIUM VALVE SPRING STEEL. (NOT SHOT-PEENED)

Fig. 13 – Fatigue and heat set data

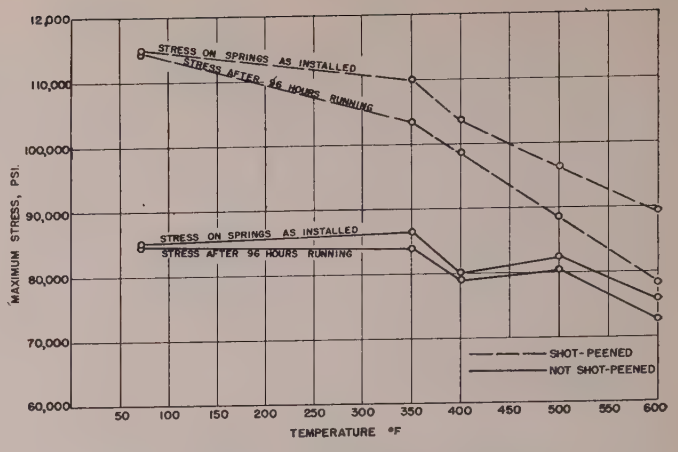

Fig. 14 – Operating stresses and temperatures permissible in 18-8 stainless valve spring steel for infinite load applications

frequently. Check runs were made on most points which seemed out of line, and in the majority of cases they were smoothed out. No check runs, however, were made in the music-wire series.

Some explanation should be suggested for the increase in endurance limit at higher temperatures noted in the case of the unpeened springs made from pretempered wire. The unpeened springs made from cold-drawn wire (music wire and 18-8 stainless steel) did not show an increase in endurance limit, as was the case with the pretempered wire, and it may be assumed therefore that similar conditions did not exist.

Since fatigue failure starts at a point of stress concentration, it is postulated that at elevated temperatures there is a tendency for stress relief or stress equalization. If a notch effect existed, it would undoubtedly overshadow the degree of stress relief. The surface condition of all the unpeened springs was essentially the same, without any notch effects, so that failure, when it started at room temperature, would occur at random positions where stress concentration developed. It was noted during all the tests that failures did not occur at the same location on the springs. The stress concentration preceding the beginning of a failure could be due to some local strain hardening caused by the repeated compressions of the springs or by inhomogeneities in surface conditions and chemical composition.

It is known that when a cold-worked metal is heated in the temperature range below the recrystallization temperature, a process of recovery develops during which surface stresses are relieved. Other effects are also observed, such as changes in electrical conductivity, but the most important change is stress relief. The increase in endurance limit on the unpeened springs was not marked until a temperature of at least 250 F was reached. Above 250 F all unpeened springs made from pretempered wire showed an increase in stress sustained.

It is postulated that above 250 F the recovery effect is strong enough to relieve stress concentra-

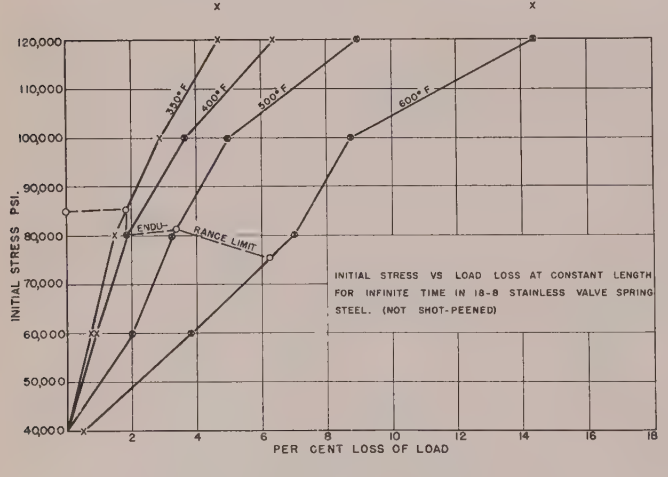
INITIAL STRESS PSI.
PER CENT LOSS OF LOAD
330° F
400° F
500° F
600° F
ENDU-RANCE LIMIT
INITIAL STRESS VS LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME IN 18-8 STAINLESS VALVE SPRING STEEL. (NOT SHOT-PEENED)

Fig. 15 – Fatigue and heat set data

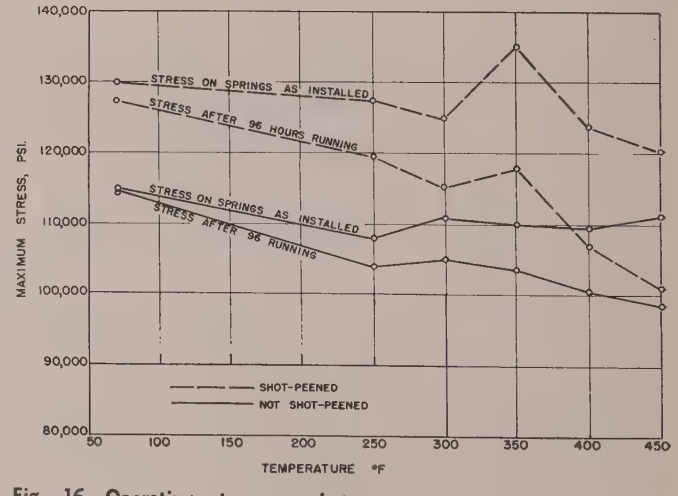

Fig. 16 – Operating stresses and temperatures permissible in music wire for infinite load applications

Table 2 – Endurance Runs at –75 F

	Shot Peened		Unpeened	
	Maximum Stress, psi	Loss of Load, %	Maximum Stress, psi	Loss of Load, %
Carbon Valve Spring Steel	135,000	1.0	95,000	<0.1
Chrome-Silicon Valve Spring Steel	135,000	<0.1	95,000	.....
High-Manganese Valve Spring Steel	140,000	1.47	100,000	<0.1
18 Cr – 8 Ni Stainless Valve Spring Steel	105,000	<0.6	85,000	<0.1
Chrome-Vanadium Valve Spring Steel	135,000	1.53	Not run	.....
High-Speed Valve Spring Steel	100,000	0.0	Not run	.....
Music Wire	Not run	.....	Not run	.....

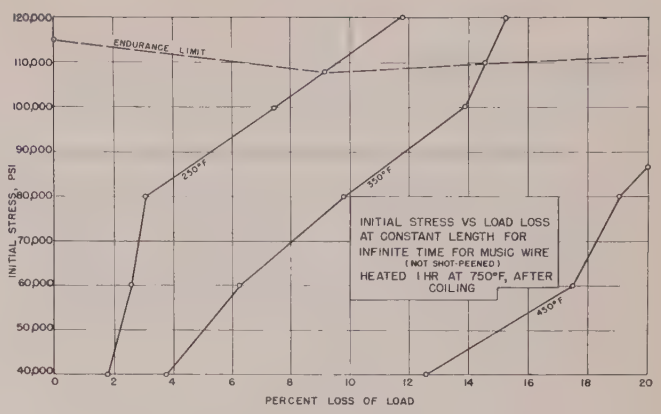
ENDURANCE LIMIT
250°F
350°F
450°F
INITIAL STRESS VS LOAD LOSS AT CONSTANT LENGTH FOR INFINITE TIME FOR MUSIC WIRE (NOT SHOT-PEENED) HEATED 1 HR AT 750°F, AFTER COILING
INITIAL STRESS, PSI
PERCENT LOSS OF LOAD

Fig. 17 – Heat set data

tions which may start to develop. This effect increases until a temperature is reached where the decrease in yield point in torsion begins to overshadow the stress-relieving effect and the endurance limit decreases.

Another possible explanation for the increase in endurance limit of the unpeened springs may be that they are in the range of blue brittleness. It might be expected that if this were true more brittle failures would have occurred in the higher temperature range (350-650 F), but in this range most of the failures recorded were due to excessive setting of the springs, which would imply softening.

In the case of the materials which are in a state of high initial stress, for example, the music wire and the 18-8 stainless steel, it is probable that less concentration at or near the surface is caused by the repeated compressions of the springs, since the existing stresses are considerably higher and are well distributed. Any stress relief due to the increased temperature would occur throughout the section as well as on the surface, and failure would therefore not start as a result of stress concentration due to the compressions of the springs. No data were found supporting this idea as far as springs are concerned, but it has been reported that the proportional limit of cold-rolled steel in tension remains fairly constant over a considerable range of temperature above atmospheric. A similar condition might exist in torsion.

The unpeened high-speed steel shows a slight rising tendency above 400 F, the endurance limit at 650 F with the modulus correction being only 2500 psi below that at room temperature.

Results and Discussion: Springs at –75 F

Table 2 includes the results of endurance runs on the various steels as far as they were completed. In most cases there is not much difference between the endurance limit at room temperature and at –75 F. Undoubtedly the reheat-treated high-speed-steel shot-peened springs would have sustained a higher stress than 100,000 psi.

The outstanding results of the low-temperature runs were the very small load losses recorded. No static load-loss tests were made at the low temperature, but it would be expected that they would have indicated proportionally low losses.

The beneficial effect of shot peening was unchanged at the low temperature, and this, coupled with the low load losses, would indicate even more satisfactory performance of shot-peened springs at the low temperatures than at atmospheric.

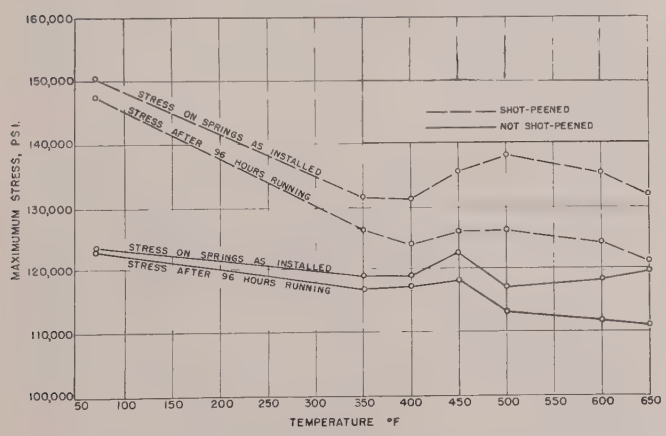

Fig. 18 – Operating stresses and temperatures permissible in high-speed valve spring steel for infinite load applications

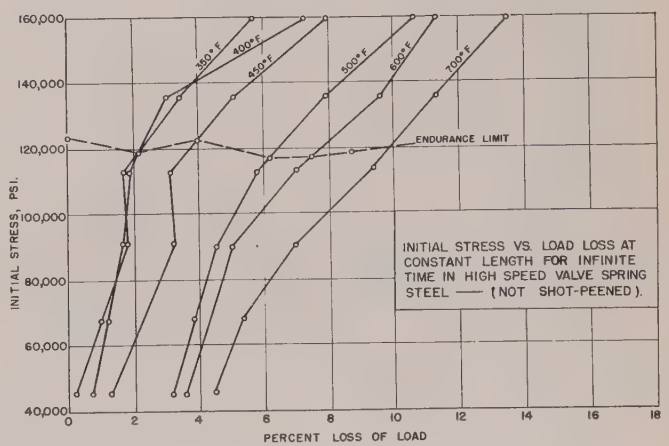

Fig. 19 – Fatigue and heat set data

Whither the

SEVERAL 2-stroke-cycle diesel engines of European manufacture are described in this paper.

The 2-stroke-cycle diesel was late to appear in the automotive scene in Europe, but noteworthy developments have taken place recently, according to the authors. In England, a couple of successful uniflow engines were placed on the market, while on the Continent the loop-scavenged engine is receiving most of the attention.

The authors feel that this is not a step backward dictated by European poverty, but perhaps a step forward toward compactness and simplicity.

THE 4-stroke-cycle automotive diesel engine early made its appearance as a vehicle powerplant in Europe. Its development reached a state where no novel significant features have appeared since World War II. The availability in Britain and Germany of many excellent 4-stroke-cycle engines capable of filling the demand for economical power in the bus, truck, and tractor field retarded the development of the 2-stroke-cycle engines. The 2-stroke-cycle automotive diesel was late in coming to the scene in Europe, more so than in the United States. Several sporadic attempts were made, but until recently they were not accompanied with commercial success.

The position of the 4-stroke-cycle engine was not challenged by the successful development of some small 2-stroke-cycle opposed-piston engines like the German Junkers, the Swiss Sulzer, and the French MAP, or by the attempts at the sleeve-valve engines which failed to reach production stage. Following American lead it was the poppet-exhaust-valve engine which first achieved commercial success by Foden in England.

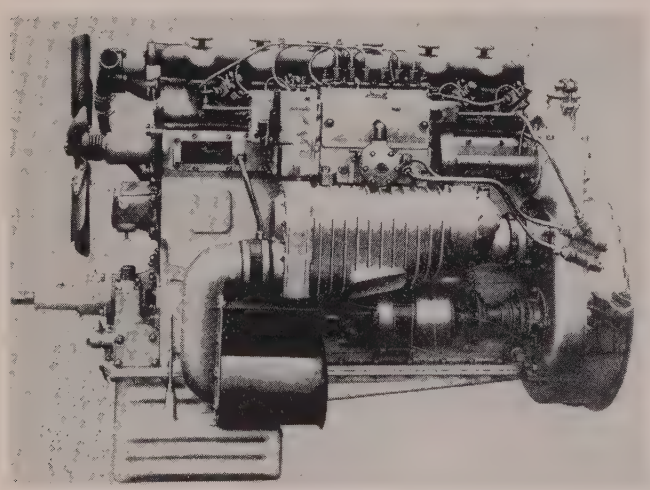

Fig. 1 – Elevation of Foden engine

Foden

The 6-cyl version of the Foden engine is shown in Fig. 1 and its transverse section in Fig. 2. It has 3.35-in. bore, 4.73-in. stroke, and develops 126 bhp at 2000 rpm, corresponding to a bmep of 100 psi. The overall dimensions are 55¾ in. long, 29⅞ in. wide, and 39¼ in. high, and the engine weighs, without electrical equipment, 1100 lb or 8.73 lb per hp.

There is nothing revolutionary in the design of the Foden engine, but every device for weight saving has been used which was compatible with sound engineering practice. Two poppet exhaust valves are in the head, and a row of rectangular inlet ports are in the cylinder. Credit is given to the Kadenacy system for the good performance of the engine, but no radical departure is noticeable from the conventional. Of course, every well-designed 2-stroke-cycle engine enjoys to a certain extent inertia charging, frequently called the Kadenacy effect. The Foden engine is certainly well designed.

The cylinder block and crankcase are one-piece aluminum-alloy castings, the dividing line being 4⅝ in. below the crankshaft centerline. The top portion of the block contains the water jacket which extends down to the inlet ports but not below. The middle portion forms the airbox and incorporates cast-iron guide vanes for better direction of the air. The lower portion is divided vertically into six compartments by the main bearing support webs. The main bearing caps are of aluminum alloy with through bolts fitted through to the

European Automotive Diesel?

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Pennsylvania State College Caterpillar Tractor Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 11, 1953.

cylinder head, which is a nickel-iron casting.

The wet cylinder liners are centrifugally cast iron and are sealed for water with two rubber sealing rings above the inlet ports and for air with a single O-ring off the crankcase. Eight rectangular inlet ports are cast in to a tolerance of 0.015 in. in the upper edge, according to claims. They have a total area of 4 sq in. and impart a tangential swirl to the air.

The main journals are 3 in. in diameter, 1½ in. long, except the rear journal which is 3 in. long. The crankpins are 2⅝ in. in diameter and 2 in. long. The gear case is at the rear, and a spiral gear splined to the front end of the crankshaft drives the oil pump, water pump, and hydraulic-brake booster pump.

The drilled connecting rod has a diagonally split big end and a stepped wristpin bearing, corresponding to stepped wristpin bosses in the piston. This arrangement increases the loaded bearing areas, inasmuch as the load in the piston is always downward. The nitrided wristpin of 1⅝ in. diameter is a press fit in the piston. The steel-backed main and connecting-rod bearing shells have white metal lining.

The piston is the most critical part of a high-speed 2-stroke-cycle engine. The Foden piston is of cast iron, is fairly long, and carries on top a fire ring, two wedge-shaped compression rings, and on the bottom a stepped air seal ring and a slotted oil control ring. The rings are prevented from rotating by snugs except the two-piece fire ring which is free to rotate with ends relieved and sufficiently chamfered to pass the ports. The piston is cam ground, oval and tapered.

In the piston crown is a toroidal cavity forming the combustion chamber into which a spray from a side-mounted single-hole injection nozzle is directed downstream, tangential to a ¼-in. diameter circle struck from the center of the piston crown. A single-hole nozzle with a relatively large (0.020 in.) orifice is, of course, an advantage in small engines as it is less likely to clog up.

The camshaft is placed high and actuates the forked exhaust-valve rockers through short push-rods. The valve throat diameter is 1¼ in., and 30-deg seats are used. Fast valve opening is essential to reduce the scavenge pressure and to take advantage of the Kadenacy effect.

The airbox pressure is only 4.5 psi with which an air delivery of 180% displacement volume is claimed, which is excellent and results in the unusually low exhaust temperature of 450 F at full load. Air is furnished by an angularly located two-lobe Roots blower which runs at 2/1 ratio and

The Authors

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C. G. A. ROSEN (M '37) is consulting engineer, Administrative Department, Caterpillar Tractor Co. He graduated from Cogswell Polytechnic College and the University of California with B.S. and M.E degrees. Mr. Rosen was vice-president representing the SAE Diesel-Engine Activity in 1940. He has also been a member of the SAE War Engineering Board.

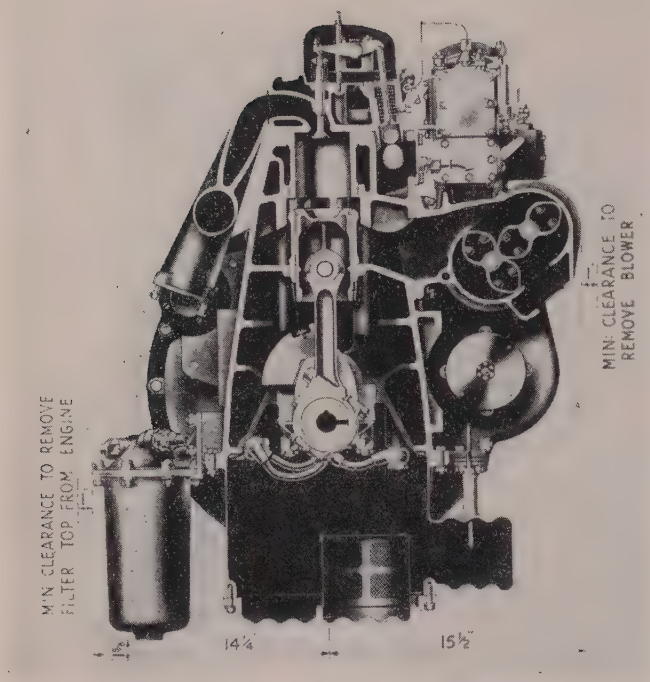

Fig. 2 – Transverse section of Foden engine

absorbs only 15 hp at 2000 rpm at full load. Significantly both the airbox pressure and the blower horsepower go up at part load, and at motoring they reach 6 psi and 20 hp respectively. This is the paradoxical result of the Kadenacy effect.

Some work has been reported on reducing the air charge at part load by bleeding air from the airbox until the airbox pressure falls to 1 psi and the blower horsepower to 10.

A typical maximum load oscillogram at 2200 rpm is revealed in Fig. 3. This shows the exceedingly smooth combustion for which this engine is noted. With an 18-deg advance using a 7-mm fuel-pump plunger and a fast cam, a peak pressure of 1650 psi is reached 2 deg atdc, with a compression pressure of 900 psi. At 350-rpm idling speed the engine ticks away like a good gasoline engine.

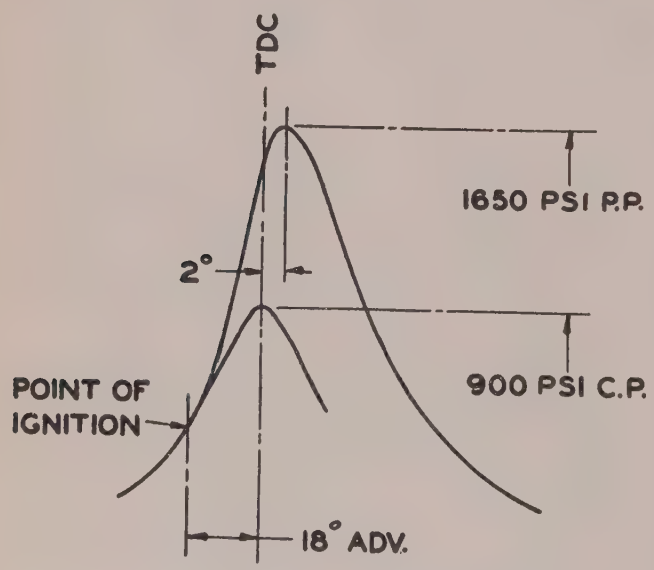

Fig. 3 – Typical oscillogram of Foden-type combustion

Fig. 4 shows the performance of the Foden FD6 engine at rated load. An economy run of a 22-ton truck gave with the Foden engine a fuel mileage of 252 ton-miles per imperial gal or 210 ton-miles per U. S. gal, at 21 mph. A 4-cyl engine of 84 hp is also built.

Südwerke

Krupp-Südwerke in Essen, Germany, builds a series of uniflow-scavenged engines with 115-mm (4.53-in.) bore and 140-mm (5.52-in.) stroke and 1700 maximum rpm. The rated horsepower is 75 for the 2-cyl, 110 for the 3-, 145 for 4-, 210 for 6-, and 300 for the 8-cyl model, but the first and the last are not being built.

As seen from the cross-section, Fig. 5, the Südwerke engine is not unlike the General Motors model 71. The view of the 4-cyl model is shown in Fig. 6.

The cylinder block is of cast iron; the cylinder head, piston, Roots blower, and oil pan are of aluminum alloy. The wet sleeves are cooled down to the oblong inlet ports but not below. The camshaft, the blower, the exhaust manifold, and the starter are mounted on the left side; the injection pump, generator, and water pump on the right side of the block. The injectors are in the center of the head and operate at 2800 psi injection pressure. The Roots blower, as well as the other accessories, are driven by V-belts.

The bearings are lead bronze, the gear-driven lubricating pump is in the oil pan but not submerged in the sump. From the performance curve

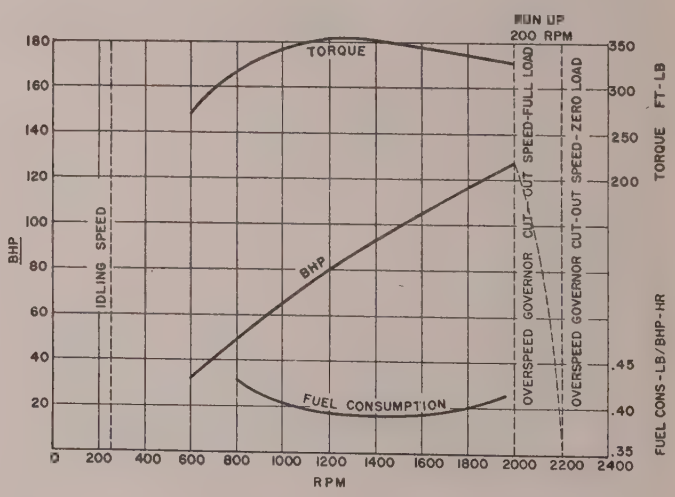

Fig. 4 – Performance curves of Foden engine

of the 4-cyl model in Fig. 7, it is seen that the full power bmep is 95.3 psi and the maximum torque bmep is 105 psi. The torque curve shows good "elasticity." The fuel consumption is fairly low. Attention is called to the different character of fuel-consumption curve of the Südwerke engine as compared with the Foden engine. The relative flatness of the Foden fuel-consumption curve down to low speeds is achieved by superior directed and organized turbulence.

Coventry

Developed for the British Admiralty, the Coventry is a small uniflow engine, 2⅞-in. bore, 4⅛-in. stroke, 2000 rpm, rated at 50 hp, with a single exhaust valve in the head, a singularly dished piston, which accommodates the spray of the side-mounted injection nozzle. A transverse cross-section is shown in Fig. 8, and the pump side is in photograph in Fig. 9.

The engine is notable for its compactness, lightness (marine), and high torque at low rpm. It will be noted that the cylinders have no liners, but the block has chrome-plated bores. The pistons are fairly long and carry wedge-type (keystone) compression rings. It has a one-piece cylinder head, a

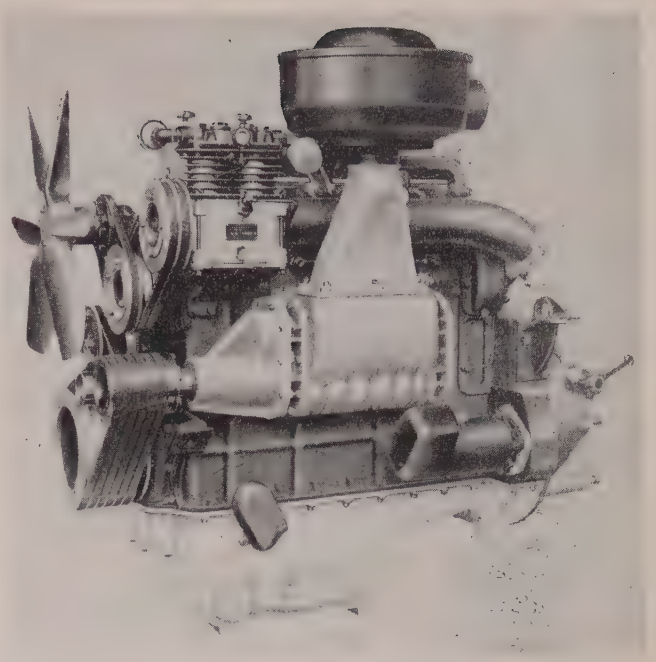

Fig. 6 – Elevation of Südwerke engine

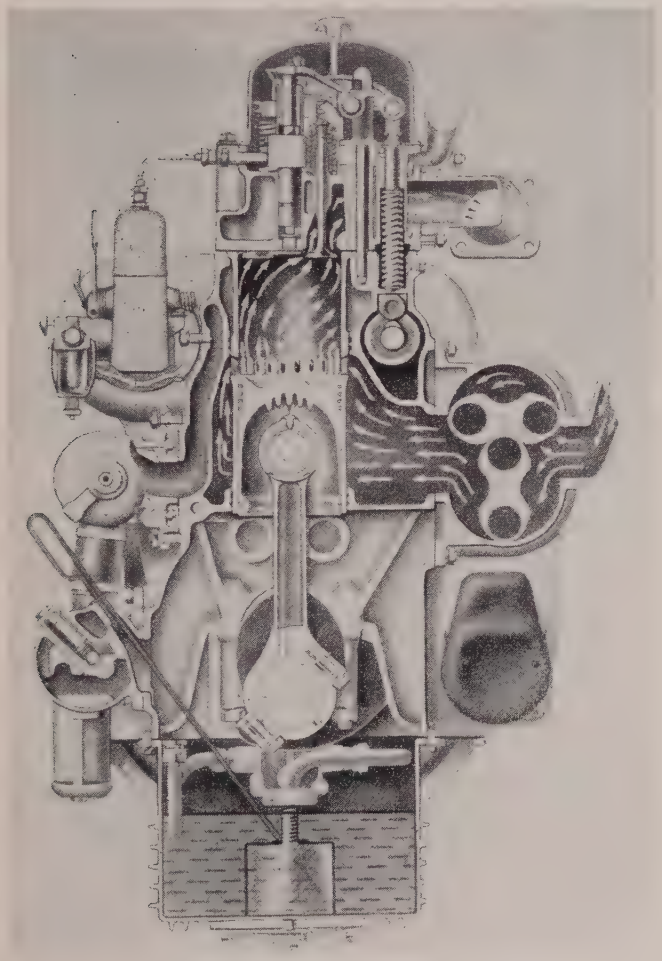

Fig. 5 – Cross-section of Südwerke engine

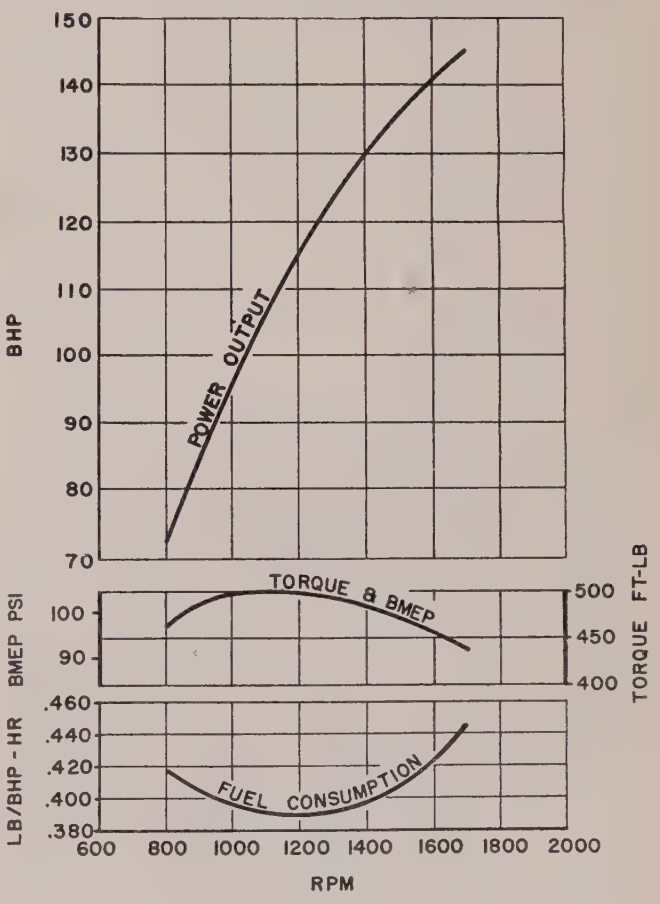

Fig. 7 – Performance curves of Südwerke engine

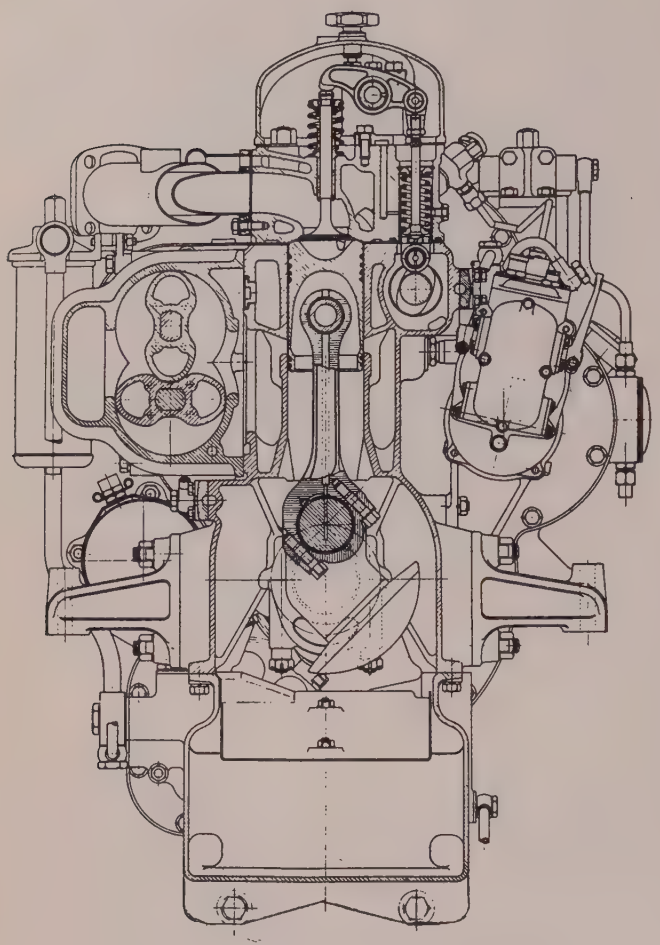

Fig. 8 – Cross-section of Coventry engine

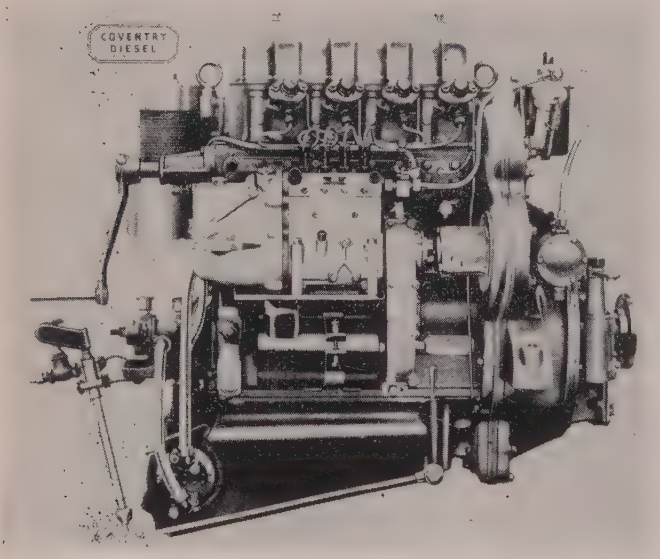

Fig. 9 – Elevation of Coventry engine

2-lobe Roots blower, a CAV injection pump, high camshaft with short push-rods. The oil cooler is submerged in the deep end of the sump, while the lubrication pump is on the shallow end.

The performance of the engine is shown in Fig. 10. The maximum torque is developed at 1400 rpm with 101.5 psi bmep. The torque curve is favorable, and the exhaust temperature is only 660 F at full load and full speed. The fuel-consumption curve exhibits good characteristics for such a small-bore engine where directed turbulence becomes more difficult to prevent the fuel spray from striking the cylinder wall and yet not causing high heat losses.

Klöckner-Humboldt-Deutz

The small loop-scavenged diesel is a postwar development, but its origin can be traced to the war years. In the search for a light, economical aircraft engine, the Nazi government enlisted Schnürle, the noted inventor of the reversed-loop or back-flow scavenging, and the resources of the Klöckner-Humboldt-Deutz concern for such a development. Under Schnürle, K-H-D assembled a talented staff, and work began in Oberursel in 1941. The combination of Schnürle's genius, the able team, K-H-D resources, and relatively unlimited German government money resulted in three years in the engine shown in Fig. 11.

This was a 16-cyl horizontally opposed engine, 6.3-in. bore and stroke, rated 1760 hp at 2670 rpm,

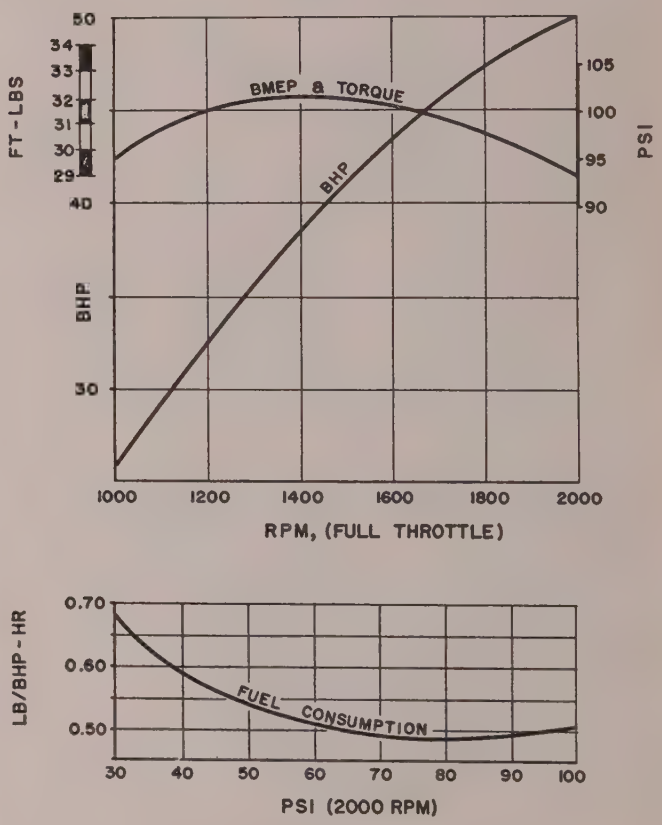

Fig. 10 – Performance of Coventry engine

with a take-off output of 2300 hp. Fig. 11 shows the engine mounted on a torque stand and Fig. 12 its measured performance, which is truly phenomenal. The specific output was 0.56 hp per cu in., remarkable for such a large bore engine; the maximum bmep well exceeded 100 psi; and fuel consumption varied gently from 0.34 to 0.4 lb per hp-hr over a wide speed and load range.

The tortoise-shell type of fuel-consumption curve is a most expressive manner in which to exhibit the character of combustion of a diesel engine. Each contour line encloses an area over which combustion performance is uniform. The flatness of profile or spread of contour is a measure of the uniformity of combustion control achieved throughout the engine operating range. A steep mountain peak type of profile would show undesirable performance. A turbocharged version was also contemplated.

Patton's invasion troops put an end to this development. Subsequently, the Oberursel plant was dismantled, the engine and what was left of the testing equipment was shipped to the United States, while the team of able engineers with the collected experience broke into groups and dispersed.

Schnürle kept a small group which in Stuttgart developed the Krauss-Maffei engine. List also acquired a small group which in Graz developed the Gräf & Stift engine; others were picked up by the United States Air Force and brought to the United States.

Fig. 11 – Klöckner-Humboldt-Deutz aircraft diesel engine

Krauss-Maffei

The Krauss-Maffei engine is shown in Fig. 13 and its transverse section in Fig. 14. This design was achieved by the collaborators of Schnürle – Otto Elwert, chief designer, and G. Emele, chief development engineer.

In four cylinders, 4.73 in. x 5.2 in., it develops 145 continuous net hp at 2200 rpm, or, intermittently, 160 hp at 2300 rpm. The overall dimensions are 35 in. long, 36 in. wide, and 44 in. high; the weight is, with all standard accessories, 1579 lb, or 10.9 lb per hp.

The 4 cyl are arranged in 90-deg V which gives evenly spaced firing comparable to an 8-cyl, 4-stroke engine. This design provides a number of distinct advantages. The generous cylinder spacing permits liberal porting with adequate directives for efficient scavenging without sacrificing engine vol-

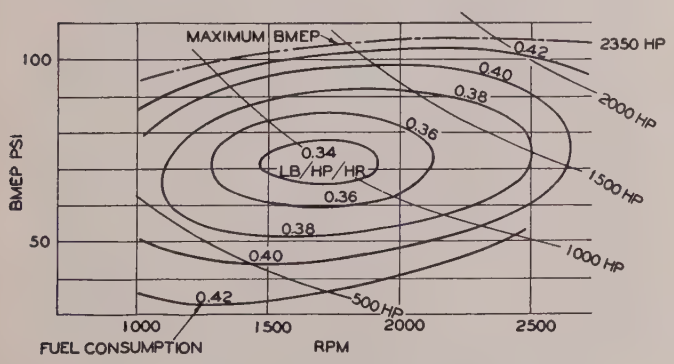

Fig 12 – Performance of Klöckner-Humboldt-Deutz engine

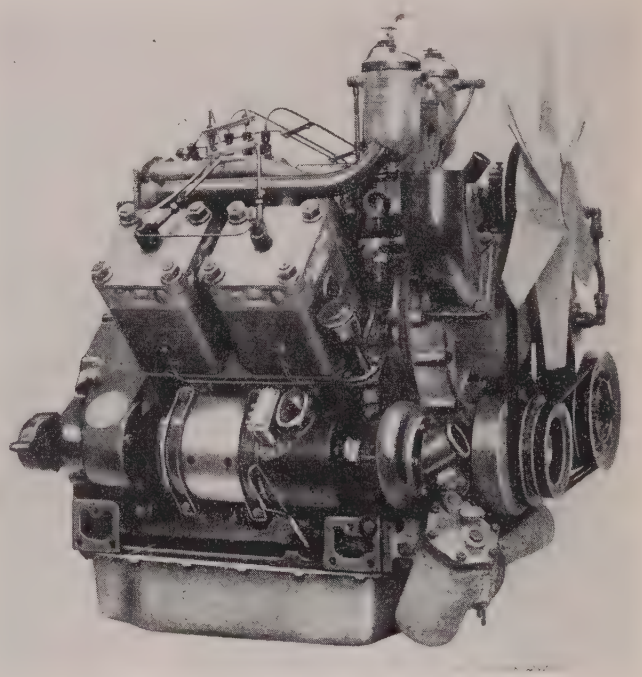

Fig. 13 – Elevation of Krauss-Maffei engine

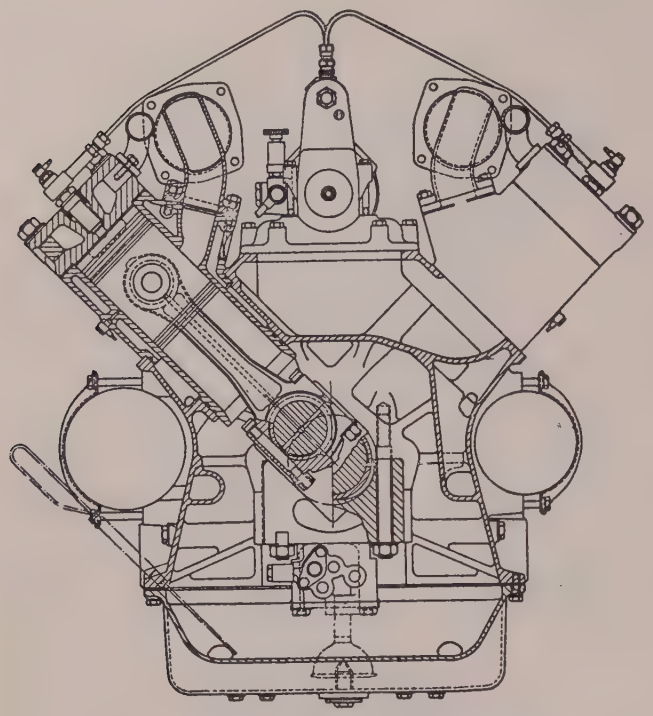

Fig. 14 – Cross-section of Krauss-Maffei engine

ume or weight. The crankshaft is short and, therefore, less susceptible to torsional problems and achieves generous bearing areas.

Furthermore, the overall benefit of this construction favors low manufacturing cost by uniformity of casting sections and simple conventional machining methods. A test case at hand involves two engines of equal horsepower and speed. The 2-stroke-cycle loop-scavenged design was listed at a price approximately 60% of that of a conventional 4-stroke-cycle engine.

The crankcase is a one-piece casting of iron to which an aluminum oil pan is attached. In Fig. 15, *A* indicates the passages from blower to receiver, *B* the air receiver, and *C* the delivery ducts to the cylinders. The two upper surfaces accommodate the cylinders, each of which is held with four studs which also secure the cylinder head and transmit the combustion forces directly to the well-ribbed crankcase. The upper portion of the crankcase serves as the airbox or scavenging receiver, closed with a cast cover which carries the injection pump. To the front the aluminum blower casing is attached, the diffuser of which connects with the airbox. No counterweights are used on the two-throw crankshaft, but one of the crankwebs carries a gear for the accessory drives. All three main bearings are 4 in. in diameter and 2 in. long. The side-by-side connecting-rod bearings are 3⅛ in. in diameter and 1⅞ in. long. This results in a very short engine (35 in. long) in spite of the generous center distance (1.85 x bore) between cylinders.

The cylinders are individual castings provided with integral cooling jackets. The lower part of the cooling jacket is formed by the crankcase. Both air and water enter the cylinders through their bottom seats. This cylinder design greatly facilitates assembly and dismantling and permits inexpensive replacement of worn components. It also gives the air the desired upward sweep so important for effective scavenging without short-circuiting.

A unique feature of the loop-scavenged engine is its extraordinary simple cylinder head. The Krauss-Maffei head is a silicon-aluminum rectangular casting with a central hole for the injector and a cavity for water around it. Furthermore, the engine needs no cylinder-head cover as there

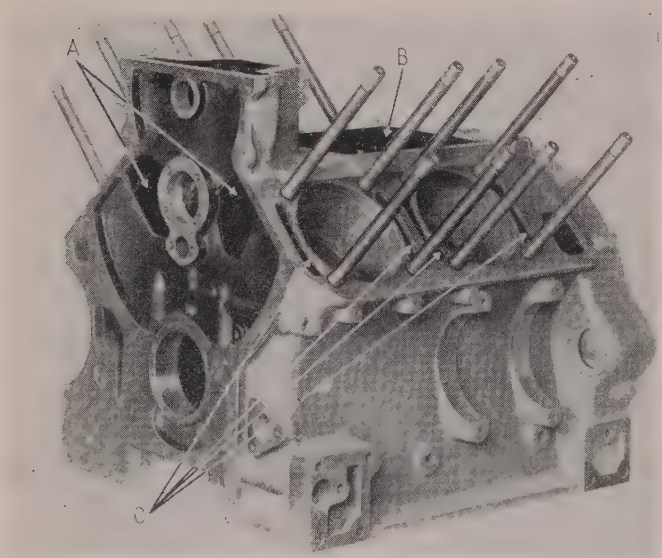

Fig. 15 – Crankcase of Krauss-Maffei engine

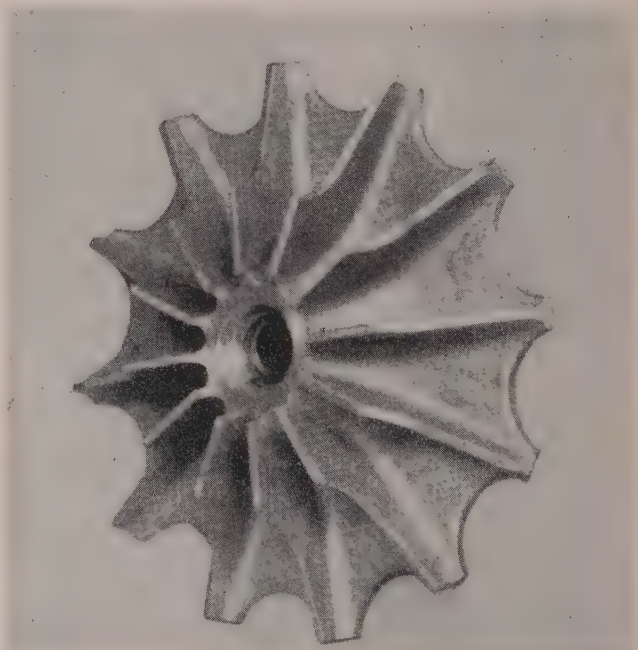

Fig. 16 – Impeller for geared centrifugal blower of Krauss-Maffei engine

is no oil splash or leak above the cylinder heads.

Scavenging air is furnished by a geared centrifugal blower, the silicon-aluminum impeller of which is shown in Fig. 16. The airbox pressure is remarkably low, 3.7 psi at 2200 rpm and 1.7 psi at 1500 rpm, which is responsible for the low parasitic horsepower absorbed by the blower and has, no doubt, something to do with the extraordinary fuel economy of the Krauss-Maffei engine.

The fuel consumption against bmep is shown in Fig. 17, indicating a minimum fuel consumption of 0.343 lb per bhp-hr at 1200 rpm and 59 bmep. At full load and top speed the fuel consumption is 0.38 lb per bhp-hr. Again the flat fuel-consumption curve with low minimum values attests to the high efficiency of mixture formation. The power and torque at full load and peak load are shown in Fig. 18.

Still more astonishing is the road consumption with this engine. One of the authors has measured 291 ton-miles per gal at 21 mph vehicle speed in Germany, and this fuel mileage has since been duplicated in this country.

Gräf & Stift

The similarity between the Gräf & Stift and the Krauss-Maffei engines is so pronounced that it could only be explained by their designers' having the same background and experience. This is indeed the case. Under the direction of Hans List this engine was designed by H. Kremser and Andreas Scheiterlein using the precious experience

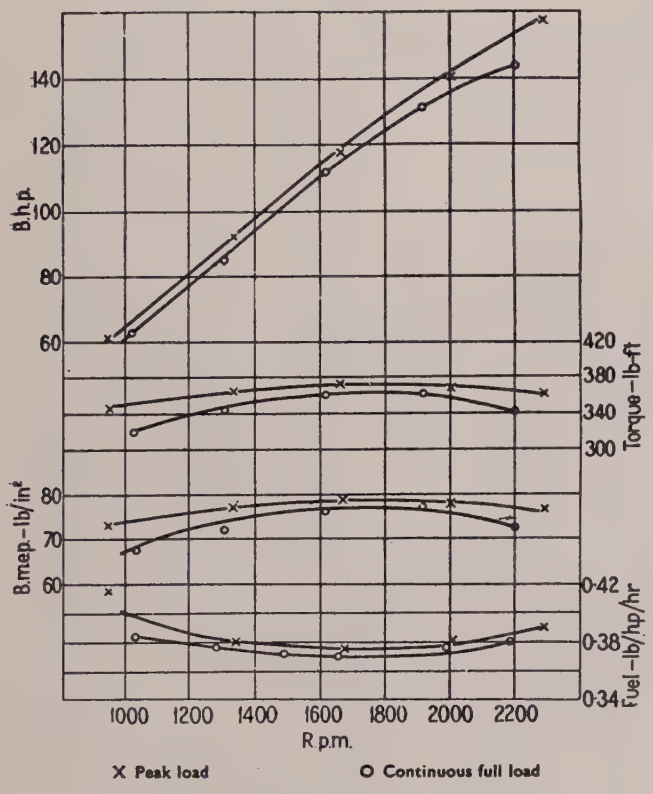

Fig. 18 – Performance curves of Krauss-Maffei engine

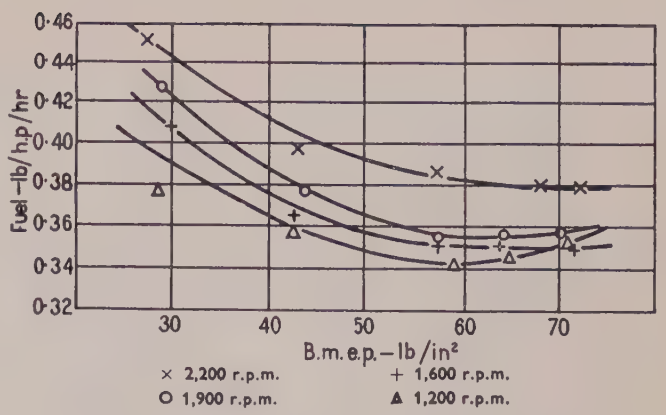

Fig. 17 – Fuel-consumption characteristics of Krauss-Maffei engine

acquired together at Deutz in the DZ 710 development.

As seen from Figs. 19 and 20, the Gräf & Stift engine is also a 90-deg V, and it is loop-scavenged through piston-controlled ports, without valves, having side-by-side connecting rods, a short, husky crankshaft, simple symmetrical cylinder head with central injector injecting into an open but dished combustion chamber. It also has 120-mm (4.73-in.) bore, but the stroke is 10 mm longer (5.51 in.), or a stroke/bore ratio of 1.16/1.

The principal differences are the following:

The Gräf & Stift has a one-piece aluminum cylinder block into which individual wet cast-iron liners are inserted. The block is carried up to the cylinder head, but it is split at crankshaft centerline, and a separate aluminum crankcase is bolted to it under which is attached the oil pan. To the

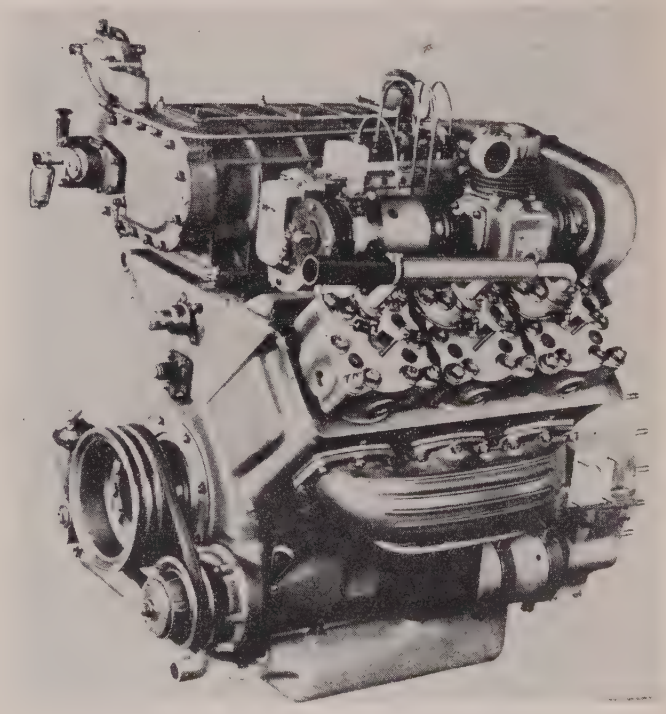

Fig. 19 – Elevation of Gräf & Stift engine

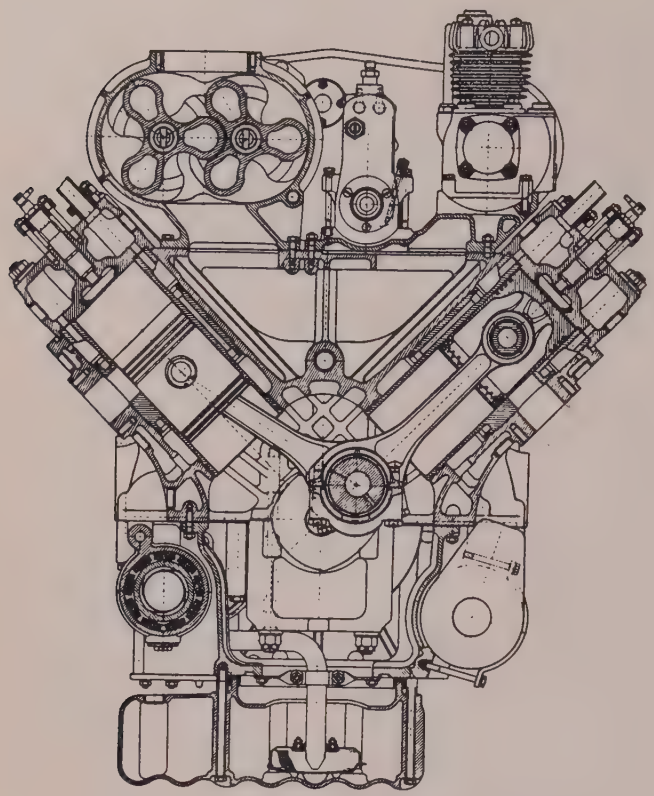

Fig. 20 – Cross-section of Gräf & Stift engine

cylinder head is bolted the liner which just hangs down and takes no part in the play of forces between crankshaft and cylinder head. Consequently, gasket compression under the cylinder head does not change the clearance between piston and cylinder head.

The dish-shaped combustion chamber is in the cylinder head and is lined with a cast-in heat-resisting steel. The piston top is completely flat, which gives the smallest surface for the heat to penetrate into it.

The Gräf & Stift cylinder liner has 10 inlet and three exhaust ports, all machined. The inlet ports communicate to the airbox in the crotch, the exhaust ports to the exhaust manifold on the opposite side. The air is supplied by a three-lobe Roots blower of liberal dimensions as seen in Fig. 19, driven at 1.51/1 ratio through a damped elastic coupling from the flywheel end of the crankshaft. Attached to the blower is a fuel supply pump. Conventional splash lubrication is used.

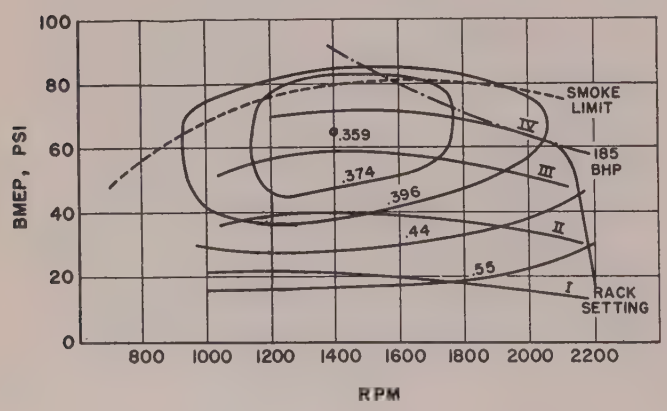

Fig. 21 – Performance curves for Gräf & Stift engine

The cooling water is supplied by a belt-driven centrifugal pump in two streams, one stream going directly to the cylinders surrounding the wet liners, and the other through the oil cooler.

The Gräf & Stift engine is being built in Vienna, Austria, in 4- and 6-cyl sizes. A performance curve of the 6-cyl engine is shown in Fig. 21. It has a displacement of 580 cu in. and is continuously rated 185 hp at 2000 rpm, but it has pulled a maximum of 252 hp at 2152 rpm. The external dimensions are 47 in. high, 36.4 in. wide, and 39.6 in. long; the weight is 1530 lb dry, or 8.3 lb per hp.

The 90-deg V-type design provides excellent balance, thus giving ideal smoothness for vehicle installations. It has the advantage that the reciprocating forces and couples of first order for the two banks add up to a constant rotating force and a couple, both rotating with crankshaft speed. Forces and couples of the first order can, therefore, be completely balanced by counterweights on the crankshaft.

However, either free secondary forces or couples or both occur in all 2-stroke-cycle V-90 engines with cylinder numbers less than 12. Because of their higher frequency (twice engine speed) and small absolute magnitude, they are usually not noticeable.

A 4-cyl engine has no free couple but a free secondary force acting in the horizontal plane. The free force is represented by the horizontal component of the vector F in Fig. 22. This vector rotates at twice engine speed and is in vertical upwards position at the time the cranks are in the vertical plane. At the crank position shown in the figure, the horizontal component of the vector is zero, that is, at that instant the free secondary force is zero. After 45-deg crank rotation the vector has traveled 90 deg, which means at this crank position the free secondary force has its maximum. The magnitude can be calculated from the equation written in the figure.

In a 6-cyl engine the secondary forces are zero, but here there is a secondary couple acting again in the horizontal plane. The couple can be represented by the horizontal components of two force vectors, as indicated in Fig. 22, rotating at twice crankshaft speed. The magnitude of the couple is written in the figure. At the instant when crank 1 is in the vertical plane, the vectors are 30 deg ahead. After 30-deg crank rotation, the vectors reach the horizontal plane, that is, at that instant the couple has its maximum amplitude. To give an idea what this amounts to in the Gräf & Stift 6-cyl engine: two unbalanced weights of 1 oz each at

12-in. radius placed in the planes of the flywheel and the fan pulley respectively and running at twice engine speed would produce a couple of equal magnitude.[1]

The various performance curves shown in this paper illustrate the excellent inherent torque characteristics for vehicle-type 2-stroke-cycle engines, especially for off-the-highway installations. Torque peaks at 15% increase are revealed at lugged down speeds of 62% of full speed.

Saurer

A very promising engine still under development is the Swiss Saurer. It is a loop-scavenged in-line aluminum job with 4.24-in. bore, 5.51-in. stroke, and the type shown in two views in Figs. 23 and 24 delivers 140 hp in 4 cyl at 2000 rpm. As seen, it is equipped with a Bosch pump and with what looks like a Lysholm-type blower. A nice oil cooler is visible in front of the engine.

The dry weight of this model is given as 1190 lb, which amounts to 8.5 lb per hp and 0.43 hp per cu in. Height is 33½ in., width 27 in., and length 39 in., surely compact.

Further development of this engine is in progress. The bore is being increased to 120 mm (4¾ in.), the scavenging efficiency is raised to 92% at a delivery ratio of 1.4 with 5.7 psi scavenging pressure. For "supercharging," a rotary valve in the exhaust has been developed, which keeps the exhaust ports closed during the compression stroke after the inlet ports are closed. A 4-cyl prototype of the improved design is to deliver 180 hp or 0.465 hp per cu in.

The company was unwilling to disclose more information on this engine, which is still in the development stage, but what was disclosed looks very promising indeed.

[1] This information was furnished by Dr. A. W. Hussmann of Pennsylvania State College, a collaborator of the late Dr. Schnürle.

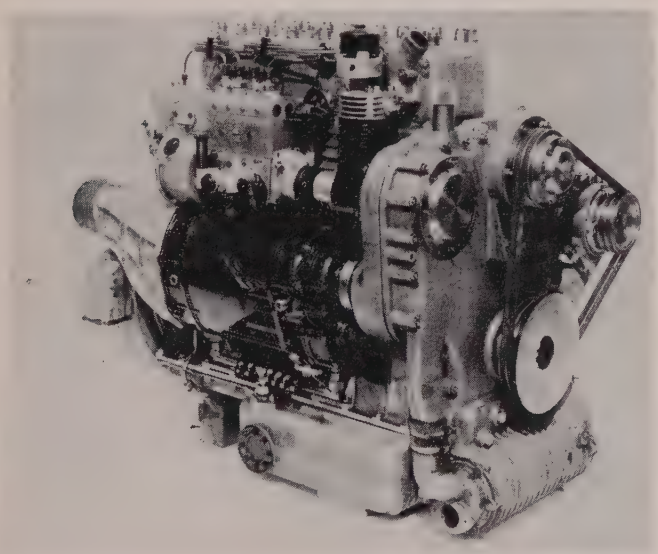

Fig. 23 – Elevation of Saurer engine

FOUR CYLINDER ENGINE

FREE FORCE WITH HORIZONTAL COMPONENT ONLY:

$$F = 2.8 \frac{W_{rec}}{g} \cdot \frac{r^2}{L} \left(\frac{\pi}{30} N\right)^2$$

F: FORCE lb
C: COUPLE lb-ft
W_{rec} RECIPROCATING WEIGHT OF ONE CYLINDER lb
g: GRAVITATIONAL CONSTANT 32.2 ft/sec²

SIX CYLINDER ENGINE

FREE COUPLE WITH HORIZONTAL COMPONENT ONLY:

$$C = 2.41 \cdot \frac{W_{rec}}{g} \cdot \frac{r^2}{L} \cdot c \left(\frac{\pi}{30} N\right)^2$$

r: CRANK RADIUS ft
L: CONNECTING ROD LENGTH ft
c: CYLINDER DISTANCE ft
N: CRANKSHAFT SPEED (RPM)

Fig. 22 – Secondary forces and couples in V-90 2-stroke-cycle engines

It appears to be rather important that the prominent Saurer Co. is seriously engaged in the development of a loop-scavenged 2-stroke-cycle engine. It will be recalled that Saurer led with the design of the toroidal combustion chamber for 4-stroke-cycle engines. This design became the model and set the trend for the subsequently popular British automotive diesel engine. What is the significance, therefore, of this new trend by Saurer in 2-stroke-cycle engines?

Atlas

A European automotive-type diesel engine used for industrial applications, that is, driving air compressors, which has enjoyed a long production rec-

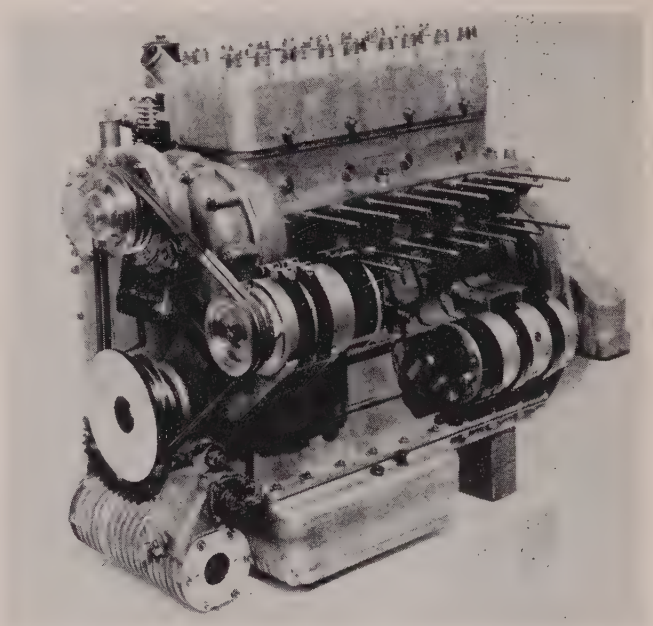

Fig. 24 – Saurer engine showing rotary valve controlled exhaust port side

ord is the Atlas diesel as manufactured in Stockholm, Sweden. This is a 4-cyl 2-stroke-cycle, loop-scavenged engine built for rugged duty and long life. It is by far the simplest design of diesel engine and illustrates the opportunities of the loop-scavenged system for low-cost manufacturing and ease of servicing. The diesel air compressor set is illustrated in Fig. 25.

The diesel engine is a 4-cyl with 3.54-in. bore and 5.12-in. stroke, operating in the governed range of 600 to 1050 rpm, depending on the blow-off of the governor. The fuel pump is mounted on the end of the crankshaft which carries the four cylinders of the diesel and the two cylinders of the air compressor. The air compressor is arranged to provide the scavenge air from the underside of the pistons. The air compressor is, therefore, of the cross-head type of construction. The starting system is very unique in that the engine is hand-started using cartridges in the cylinder head for supplying the localized temperature to achieve initial ignition. The cylinder head is of cast iron, of very simple construction with a dished dome, and has but two openings, one for the fuel valve and one for the starting plug for each cylinder. The cylinder jackets are of cast iron and the crankcase of cast-iron open construction. The crankshaft is a six-throw single-piece crank, four throws for the diesel and two throws for the air compressor.

The cross-section view in Fig. 26 shows very similar porting arrangement to that used in the Schnürle system and was developed by Herman Pyk. Each cylinder is provided with four scavenging ports and three exhaust ports. The scavenging ports are given well-guided direction to the flow of air into the cylinder. The positive-displacement piston pump supplies scavenging air from 3 in. of Hg to 9 in. of Hg, or a maximum of about 4½ psi at 1600 rpm and 80 bmep output.

The manufacturers claim a minimum fuel consumption at 0.37 lb per bhp-hr at 1400 rpm. The performance data are plotted in Fig. 27 against bmep and show scavenging air pressure characteristics. In almost all respects the combination of compressor and engine represents an excellent balanced engineering design to achieve reliability and simplicity.

The piston is of cast iron and carries the contour for the combustion chamber similar to the Mexican-hat design. There are five compression rings, which are of the simple snap-ring, lapped-end-type construction and are pinned in place to prevent snapping into the ports. There are two oil rings for oil control. The fuel valve injects directly into the cylinder and has eight 0.008-in. diameter holes forming a conical spray with an included angle of 142 deg. The governor is arranged to provide full-load engine speed when the unloader valve requires this speed for charging the receivers. At a preset air pressure unloader valve unloads the compressor and reduces the engine speed to 600 rpm idling operation.

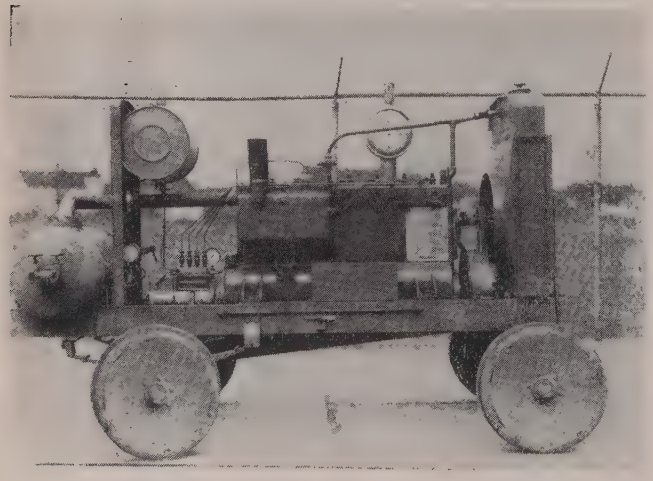

Fig. 25 – Elevation of Atlas diesel compressor set

Uniflow- Versus Loop-Scavenging

Earlier automotive 2-stroke-cycle engines had uniflow-scavenging, which is acknowledged as the most perfect scavenging system. Nevertheless, recently several small loop-scavenged engines have made their appearance. Is this a step backward dictated by European poverty?

The specific output of a 2-stroke-cycle engine is largely determined by its scavenging efficiency, which is a measure of the success in clearing the cylinder of the residual gases from the preceding cycle. The bmep of an engine can be expressed[2] as:

$$\text{Bmep} = 180 \frac{0.4}{f} \frac{14.5}{r_{th}} \frac{\eta_{sc}}{1 + (\lambda - 1)\eta_{sc}} C_{rel}$$

where:

f = Specific fuel consumption, lb per bhp-hr

r_{th} = Theoretical air/fuel ratio

λ = Excess air factor

C_{rel} = Relative charge in terms of displacement volume

The bmep increases with the relative charge and with the scavenging efficiency. If we assume $f = 0.4$, $r_{th} = 14.5$, and $\lambda = 1.6$, which are good average values, then:

$$\text{Bmep} = 180 \frac{\eta_{sc}}{1 + 0.6\eta_{sc}} C_{rel}$$

C_{rel} depends on the pressure and temperature of the charge trapped in the cylinder. Its value ordinarily is not far from 0.9 in "nonsymmetrically scavenged" engines as most uniflow engines are and 0.75 in "symmetrically scavenged" engines as most loop-scavenged engines are. When both inlet and exhaust are piston controlled we have symmetrical scavenging as the port opening angles

[2] See "Scavenging of Two-Stroke-Cycle Diesel Engines," by P. H. Schweitzer. Pub. by Macmillan, New York, 1948.

are symmetrical relative to bottom center. When the exhaust is valve controlled, it can be closed earlier and some supercharge effect achieved.

The scavenging efficiency itself varies with the delivery ratio and with the scavenging arrangement, as shown in Fig. 28. Cross-scavenge is the poorest, as considerable amount of short-circuiting is involved. Fresh air blows through the cylinder without driving out residual gases. Loop-scavenge pioneered by Schnürle and Curtis is much better, and uniflow-scavenge is the best.

With 140% delivery ratio, which is a good average figure for blower-scavenged engines, the respective bmep's are 58, 75, and 92 psi for cross-, loop-, and uniflow-scavenge, respectively.

The 20% lower bmep represents a serious disadvantage for the loop-scavenged engine. Are there enough compensations to make it attractive in spite of this shortcoming?

Let us examine first how the reduced bmep affects the weight and bulk of the loop-scavenged engine. In Table 1 the principal dimensions, weights, and power output of the engines mentioned are tabulated. The first three are uniflow-scavenged, the rest loop-scavenged. Some are aluminum, others cast iron. Some are in-line, others V-type engines.

Comparing the bmep's of these engines we find that the average for the loop-scavenged engines is indeed some 22 psi lower. But their horsepower output per cu ft space is, if anything, higher than those of the uniflow engines.

We don't have to search hard to find an explanation for this apparent anomaly. In the first place, the rotative speeds of the loop-scavenged engines are somewhat higher because of the absence of poppet valves. More important, the loop-scavenged engines have no camshafts, push-rods, rocker

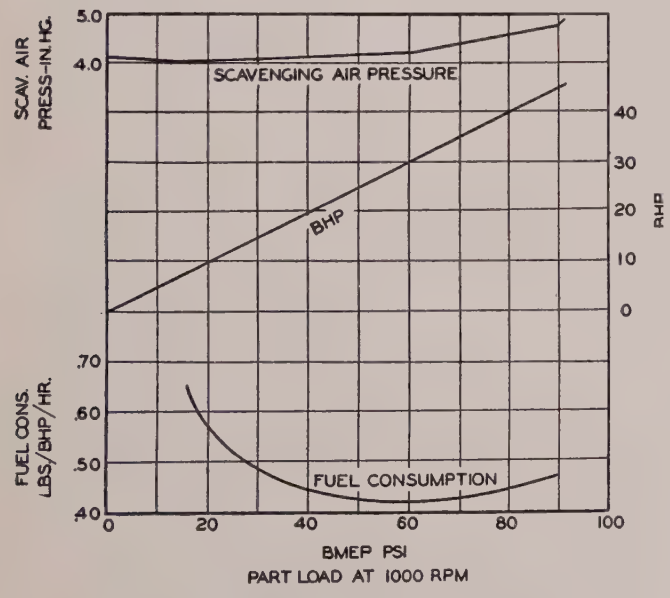

Fig. 27 – Performance curves of Atlas engine

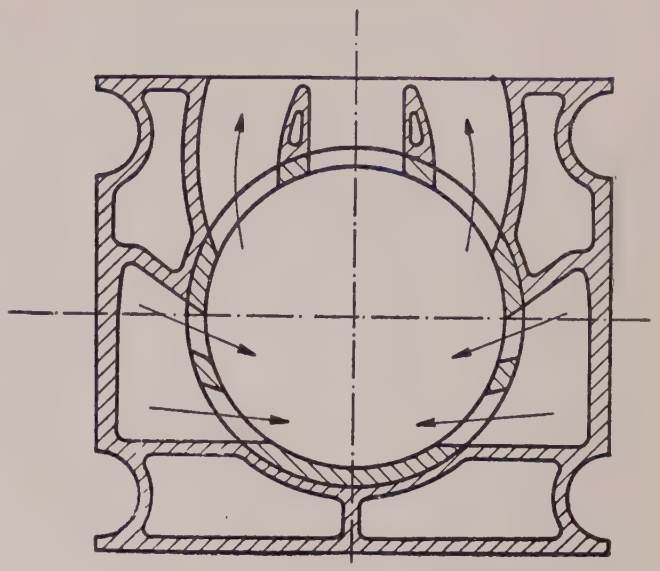

Fig. 26 – Cross-section of Atlas cylinder

arms, valves, and cylinder-head covers. These components have weight, and they increase the bulk of the engine without affecting its cylinder displacement. The result is that on the hp per cu in. displacement basis the uniflow engines are higher, but on the hp per cu ft volume basis they are lower. If we worship the bmep we should not use 2-stroke-cycle engines at all but supercharged 4-stroke-cycle engines. But the driver and the

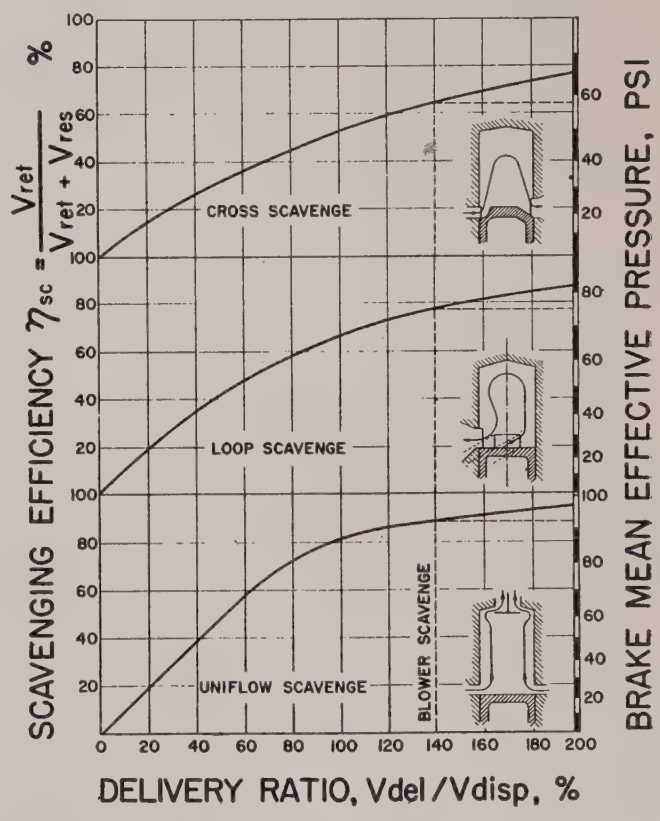

Fig. 28 – Scavenging efficiencies of 2-stroke-cycle engines

Table 1 – Principal Dimensions, Weights, and Power Outputs of Engines Discussed

	Bore and Stroke, in.	No. of Cyl	Displacement Volume, cu in.	Rated Rpm	Rated Hp (Continuous)	Bmep at Rated Rpm, psi	Maximum Bmep at Specified Rpm	Hp per Cu In. Displacement	Dry Weight, lb	Specific Weight, lb per hp	Height, Length, Width, in.	Space Requirements, cu ft	Hp per Cu Ft	Specific Fuel Consumption at Rated Hp, lb per bhp-hr	Best Fuel Consumption (Part Load), lb per bhp-hr
Foden FD6, Aluminum, Uniflow-Scavenged	3.35–4.73	6	250	2000	126	100	106 at 1200	0.505	1100[b]	8.73	39.25 55.75 29.87	37.9	3.33	0.41	0.392
Südwerke, Cast Iron, Uniflow-Scavenged	4.53–5.52	4	354	1700	145	95.3	99 at 1150	0.41	1540	10.6	46.8 45.7 28.2	35	4.15	0.44	0.395
Coventry KF, Cast Iron, Uniflow-Scavenged	2.87–4.12	4	107.1	2000	50	93	101 at 1400	0.466	622	12.4	34.12 36.12 24.25	17.4	2.87	0.49	.....
K-H-D DZ 710,[a] Loop-Scavenged	6.3–6.3	16	3150	2670	1760	83	105 at 2200	0.56	2880	1.28	37 80 53	91	19.3	0.4	0.34
Krauss-Maffei KMD6, Cast Iron, Loop-Scavenged	4.73–5.12	4V	360	2200	145	72	77 at 1700	0.403	1579	10.9	44 35 36	32	4.53	0.38	0.343
Gräf & Stift 6 ZT, Aluminum, Loop-Scavenged	4.73–5.51	6V	580	2000	185	63.2	71 at 1400	0.319	1530	8.3	47 36.4 39.6	39.2	4.72	0.39	0.36
Saurer, Aluminum, Loop-Scavenged	4.24–5.51	4	326	2000	140	85	.....	0.43	1190	8.5	33.5 39.4 27.1	21.35	6.56	.....	.....
Atlas, Iron, Loop-Scavenged	3.54–5.12	4	201	1000	40	80	85 at 1400	0.20 0.32	.....	.....	.....	.....	.....	0.47	0.41 0.37

[a] Aircraft experimental.
[b] Less electrical equipment.

vehicle owner pay little attention to the bmep. They want most power in the smallest package. Apparently the loop-scavenged engine has it.

By the same token the loop-scavenged engines have the greatest simplicity. The elimination of the valves, valve gear, and upper lubrication reduces the number of moving parts to an absolute minimum which simplifies manufacture and service, and lowers friction horsepower.

H. Desmond Carter called attention to the paradox that the engine which is by far the simplest mechanically has been the most complicated to design and develop – to which Sir Harry Ricardo replied: "The loop-scavenge diesel engine in the past has suffered terribly by persecution by the novice. . . ." He himself ". . . had not been enough of an amateur to be seduced by its apparent simplicity or enough of an expert to face its formidable problems." That was in 1946, and since then the most significant developments in small loop-scavenged engines have taken place.

Perhaps the loop-scavenged automotive diesel is not a step backward after all but a step forward toward compactness and simplicity.

DISCUSSION

Lists Other Advantages of Loop-Scavenged 2-Stroke Diesel

– P. J. Carp
Consulting Engineer

IT is certainly a great satisfaction for a European engine designer to see that the "poor man's engine" has begun to get recognition on this continent.

Although, when quoting H. D. Carter and Sir Harry Ricardo, the paper hints at the difficulties of assessing the possibilities of 2-stroke engines due to their formidable gas-dynamic problems, I do not think that enough has been said about the available experience in this field and the conclusions which can be drawn from it. Any new development would be regarded with suspicion, if experience were not available to back not only theoretical consideration but also comparisons with other types of engines. If we, therefore, want to assess properly the merits and the possibilities of the uniflow- and the loop-scavenging processes, we will have to analyze their long, mainly European, experience.

This experience goes back to the early twenties, when Junkers decided that it made no sense to have "as many scavenging pumps as one had cylinders" (the 4-stroke engine), and started quantity production of his opposed-piston uniflow-scavenged diesel, which reached the height of its development during World War II, with the Jumo 205 turbosupercharged diesel engine for strategic reconnaissance planes. Others like Sulzer, Ste. Lilloise, Pescara, Maegerle, M.A.P. started a little later. Valve-type engines were tried by Alfa-Romeo, as well as Krupp (which is in production today); furthermore, a considerable amount of research work has been done with sleeve-valve uniflow-scavenged engines by Ricardo, SNECMA, Rogifer, and with U-type cylinders by Garelli, DKW, and others.

Even more experience has been gathered with loop-scavenged engines, beginning with the early Thirties, on both gasoline and diesel engines. It is during this period that Schnürle patented his well-known scavenging process. Many motorcycle and car manufacturers, the best known of which is DKW, built hundreds of thousands of excellent Schnürle-scavenged gasoline engines and explored their ultimate possibilities in ear-splitting racing engines. During 1934, Klöckner-Humbolt-Deutz and their chief engineer, Kremser, decided that it was time to think about a cheaper diesel engine, and they developed several lines of loop-scavenged diesels, which went into production beginning with 1936. They produced during World War II a 1200-hp light marine engine, which achieved, unsupercharged and in cast-iron construction, the still incredible weight/power ratio of 13 lb per hp and specific space requirements of 29 hp per cu ft at only 700 crankshaft rpm! This result should have had, I think, an explosive impact on engine design all over the world, but, curiously enough, this was not so.

Supercharging of engines with symmetrical timing diagrams was tried as early as 1924 by a Berlin firm on its

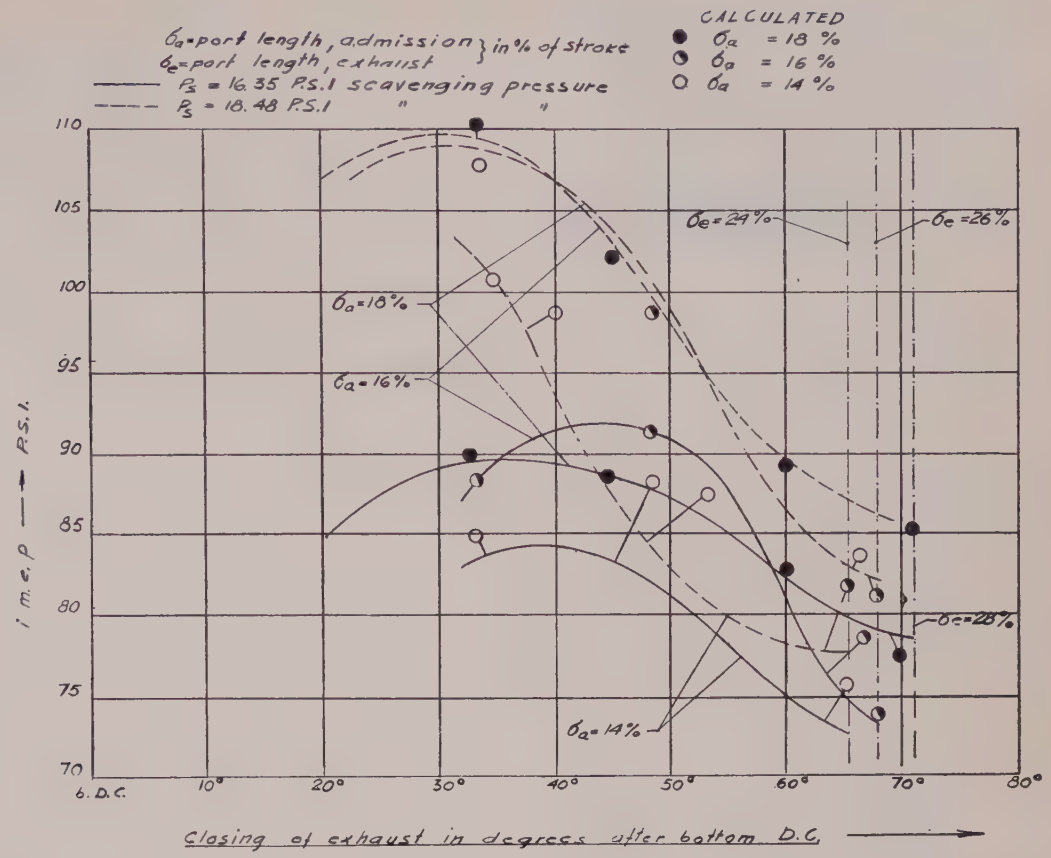

Fig. A – Indicated mean effective pressure versus exhaust closing

cross-scavenged 3-cyl 1100-cc gasoline engine. A rotary valve in the exhaust channel, directly behind the ports, was used. The same system has been successful, for many years, in the M.A.N. large double-acting ship engines.

The examples quoted above show that a great deal of experience has indeed been gained in Europe about the development of the 2-stroke engine, in which I personally was involved. This leads to a first conclusion: speaking of scavenging efficiency, the paper states that " . . . uniflow-scavenging is best . . . " Although mathematical analysis supports this statement, facts do not. We must not forget that mathematics cannot seize all the complicated phenomena and has to rely on many assumptions, which may or may not be correct. The facts show that the Schnürle-scavenged engine has the best fuel efficiency, 5 to 10% better than the uniflow-scavenged engine for similar combustion processes, similar cylinder sizes, and similar rpm's and piston speeds. The Krauss-Maffei and List-Graf & Stift engines show consumptions of 0.343 and 0.36 lb per hp-hr, against 0.392 for the best uniflow-scavenged engine men-

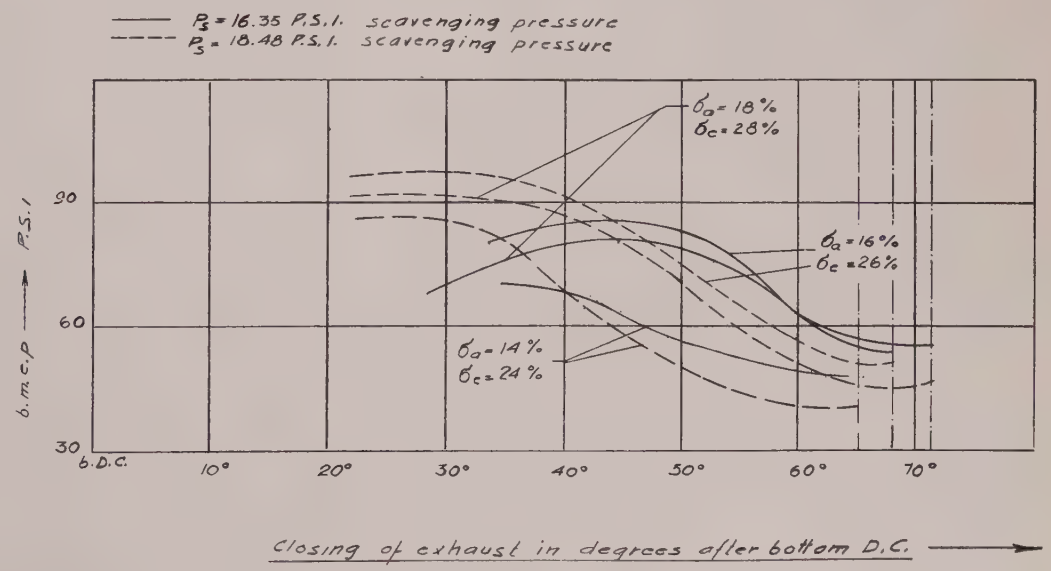

Fig. B – Brake mean effective pressure versus exhaust closing

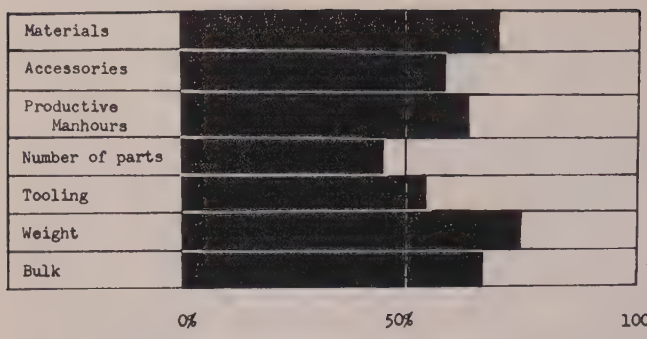

Fig. C – Comparison between AVL 170-V-6 traction engine of 500 hp at 1600 rpm with a modern supercharged V-12, 4-stroke engine of 500 hp at 1900 rpm. Piston speed of both engines is 1900 fpm. The 4-stroke design is rated at 100%

tioned in the paper. (The lower figure in the List engine is due to the use of a Roots blower with a lower adiabatic efficiency than the centrifugal blower in the Krauss-Maffei.) Similar figures were found by Veritas on the Schnürle-type Deutz small marine engine (V-8, 420 hp at 600 rpm), which belongs to a line of engines initiated before the war by Kremser, who designed after the war the List engine. This Veritas report, under the signature of Coppens, states " . . . the specific fuel consumption of this engine is the smallest found by us for engines of a swept volume of less than 12,000 cu in. per cylinder (very large ship engines) . . . " Performance facts therefore prove that the loop-scavenged engine has the best efficiency. The reason is that the scavenging air in the uniflow engine does not expel the gases in a piston-like fashion. Heavy mixing rather takes place, much more so than in the Schnürle process. Here, the air is guided by the cylinder walls around the gases, which escape by their own pressure, by volume displacement, and later by their inertia and gas-dynamic effects (Kadenacy). Little is left to "scavenging" proper by the scavenging air at the very far end of its travel.

This mixing of air and gases in the uniflow-scavenged engine is the main reason for the very low exhaust-gas temperature, mentioned in the paper, for certain diesels of this type, since this temperature is much lower than corresponding to the expansion ratio and the heat losses.

Since scavenging efficiency affects directly the mechanical efficiency, the fuel-consumption figures quoted above seem to support the fact that a good Schnürle-scavenged engine has a better efficiency than even the uncompounded 4-stroke engine, which has as many "scavenging pumps" as it has cylinders and which has to drive its valve mechanism as the 2-stroke uniflow engine does.

Our first conclusion is, therefore, that the Schnürle-type engine has not only, as the paper states, "most power in the smallest package" but best efficiency for uncompounded engines as well.

Speaking of "most power in the smallest package," we come to supercharging. Here, in principle, the basic Schnürle type, as any engine with a symmetrical timing diagram, cannot be supercharged (discounting gas-dynamic effects which make this possible to a limited degree). As we have seen, even with this apparent handicap, this engine actually has "most power in the smallest package" because the bulk and weight advantages more than compensate for the 20% lower bmep. But, this type can be supercharged. The paper has already disclosed the Saurer development with a rotary valve. This method is not new. I just mentioned the small Berlin firm of the early Twenties. I also mentioned the M.A.N. large ship diesels. List and Niedermayer of Graz have done, and still do much research and development work with this device. Under certain conditions and for unchanged scavenging pressures and therefore outlays, considerable augmentations in bmep can be obtained. (Figs. A and B show some results obtained by List and Niedermayer.) Of course, much has still to be done, but the feasibility of the principle has been established. Further, it points to the way to supercharged loop-scavenged engines without sacrificing their basic simplicity. Indeed, this rotary valve has no fit in the exhaust channel. It runs with a clearance, since it seals only against scavenging pressure, and air losses are very small and do not matter. It is just a tube, with proper ports, running at engine speed in the exhaust passage. Nothing could be basically simpler and more reliable from the mechanical point of view. Of course, the gas-dynamic problems of this whole scavenging and supercharging process are far more complicated than generally believed, but years of research have brought many positive results. In any case, the basic mechanical simplicity is there. No inertia problems limit the speed of this rotary valve, and therefore the advantage of speed-potentiality of the loop-scavenged engine is maintained, contrary to the 2-stroke valve engine, with its severe inertia and distortion problems. So, supercharging being possible, we certainly have not reached the end of the possibilities of the loop-scavenged engine.

I would like to give other advantages of the loop-scavenged engine not mentioned in the paper: First, there is more freedom in designing the combustion chamber (since the head is unobstructed by valves) so the engine can digest a wider range of fuels with high efficiency. Secondly, larger and fewer cylinders, for the same weight and bulk, produce a correspondingly better efficiency and softer combustion. Thirdly, there are better prospects of compounding (in comparison with the uniflow-scavenged valve engine) because of better impulse utilization and higher exhaust-gas temperature due to the reduced air dilution (better scavenging efficiency).

It is, therefore, quite probable that this type of engine will retain even in the future its present advantage of "most power in the smallest package." The only automotive diesel of this type in production today is the List-Graf & Stift engine, which, in accordance with the figures published in the paper, has already attained the best specific bulk figures of 4.72 hp per cu ft, the best specific weight of 8.3 lb per hp, and both at very conservative bmep's of 63.2 psi and rotational speeds of 2000 rpm, and the best fuel consumption in comparison with all uniflow engines mentioned. It is therefore easy to figure out how this advantage will grow with less conservative ratings and later with supercharging.

We, therefore, not only support the conclusion of the paper that the loop-scavenged engine had attained in its fully developed production form (Graf & Stift) the lowest fuel consumptions, specific weights, and specific space requirements, but we would like to complete it by stressing the future potentialities of this type as mentioned before.

Unfortunately the paper only hinted at the cost problem. In view of the competition of the increased efficiency of the gasoline engine and the rising diesel fuel prices, initial cost has to be closely considered, and this is in America as well as in Europe. Actually, the highly competitive United States economy should put much more emphasis on initial cost and efficiency than Europe with its many duty-protected economical spaces which often allow expensive and unefficient designs to get into production. I therefore think that the loop-scavenged diesel engine may be a European answer to an American problem.

Considering only present designs and based on similar production and financial conditions, I have prepared in Figs. C and D two graphs giving, in per cent, cost comparisons between two typical List (AVL) designs with two well-known American designs. These graphs are self-explanatory.

And looking into the not-too-distant future, I cannot see how the 4-stroke supercharged diesel can possibly compete

pricewise with the 100-psi bmep loop-scavenged engine, since 100 psi of the 2-stroke cycle is equivalent of 200 psi for the 4-stroke cycle, which cannot be obtained otherwise than by multistage supercharging with intercooling! In addition, the loop-scavenged supercharged diesel will always beat costwise its uniflow brother. These arguments may answer the questions of the paper about the "significance of the trend by Saurer," and whether the "loop-scavenged engine is a step backwards or a step forward toward compactness and simplicity."

Discusses the Case for 4-Stroke Diesel Engines

– J. H. Pitchford
Ricardo & Co., Ltd.

WHILE some reference is made in the opening paragraph to the 4-stroke automotive engine in Europe, the authors concern themselves in the body of the paper exclusively with 2-stroke developments. This approach, which is not incidentally reflected in any way in the title, can in the writer's opinion give a misleading impression not only of the present situation but also the future trend in automotive diesel technique in Europe. For every one of the 2-stroke engines named and discussed there are a hundred 4-stroke automotive engines in service in Europe today, so that numerically the picture for the present, and for the immediate future, is overwhelmingly dominated by the conventional in-line, 4-stroke engine. The output today of such makers as A.E.C., Leyland, Gardner, Birmingham & Midland Motor Omnibus Co., Bristol, Renault, Berliet, Citroen, Fiat, Lancia, Alfa-Romeo, Mercedes-Benz, Bussing, and Henschel, amount in the aggregate to a formidable figure, and the writer believes he is right in saying that there is no 2-stroke engine nor prospect of one amongst them.

There is a reason for this state of affairs of which more might have been made. In Europe today—and there is little reason to believe that the situation will change in any dramatic way in the near future – fuel consumption is all important. By fuel consumption is meant essential road fuel consumption under largely urban and suburban conditions where traffic density and builtup conditions dictate a low mean load factor for an engine which must for accelerating purposes only have a maximum power in the 100- to 130-hp range.

These are conditions of operation in which the "straight" 4-stroke engine always has a very great advantage over its 2-stroke rival on the score of fuel consumption, where both engines operate up to approximately the same maximum power: one cannot at a mean load factor of around 35% expect to have one's scavenging done for nothing, especially where a low-efficiency Roots-type scavenge blower is used.

Another point of importance in crowded urban conditions is that of exhaust cleanliness, particularly in bus service with several stops to the mile. The writer has not yet seen a 2-stroke engine of any make whose behavior as regards freedom from exhaust smoke when moving away under load after a stop of 30 sec to 1 min would be tolerated in any city bus service in this country or in France or Italy.

It is disappointing not to see in the review of European makes of automotive diesel engines any reference either to the Mercedes-Benz 108 cu in. engine or to its Italian counterpart, the Fiat 615 N. Both these are, of course, "straight" 4-cyl 4-stroke units, and they represent a very strong trend at the present time on this side of the Atlantic towards small automotive diesel engines of low road fuel consumption under urban conditions.

Some of the test-bed fuel-consumption curves shown are quite astonishingly good, but there is, it must be admitted, a very considerable discrepancy between the different makes and types of engines to which reference is made.

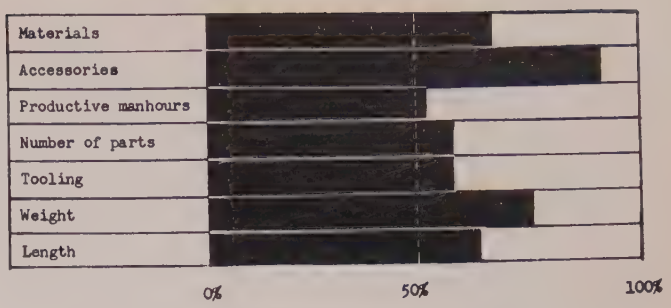

Fig. D – Comparison between an AVL 120-V-6 vehicle engine of 200 hp at piston speed of 1830 fpm with a well-known 2-stroke engine of same power and piston speed

A serious disadvantage of the loop-scavenge system for application to engines which must run up to high speeds is the fact that no organized air movement for fuel distribution can be superimposed upon that required for scavenging without serious interference with the latter function. This state of affairs can only be met, and then only to a limited extent by the use of a four- or even six-hole nozzle in an attempt to achieve acceptable fuel distribution at high speeds. The service difficulties consequent upon this technique are well known.

Larger Bore, Shorter Stroke Will Reduce Diesel Weight and Cost

– L. C. Lichty
Yale University

THE engines listed by the authors, with the exception of the aircraft engine have bore/stroke ratios appreciably less than unity and an average weight of about 10 lb per bhp. Compared to the aircraft engine weight of 1.28 lb per bhp, the weights of the other engines appear altogether too high.

The gasoline-engine trend for some time has been toward bore/stroke ratios appreciably greater than unity. When it is realized that for every inch the stroke of an engine is reduced, in general, the cylinder length is reduced 1 in., the connecting rods are reduced 2 in., and the crank circle is reduced 1 in., it can be seen that the engine height may be reduced 4 in. per in. reduction in stroke. Why do the designers of diesel engines for road vehicles still hold to their small-bore long-stroke engines when a very marked reduction in weight can be made by using shorter strokes with correspondingly longer bores which result in the same engine displacement? Is it the combustion system that is preventing them from making this change, or is it the normal resistance against any change?

A combination of the V-type arrangement of the cylinders, which some of the designers are using, and the larger than unity bore/stroke ratios should result in equal or more rigidity, much less weight, probably less friction, and less costly engines.

Authors' Closure To Discussion

The comments by Mr. Carp and his supplementary information on the recent progress of the loop-scavenged 2-stroke-cycle engine are welcome. The authors can support his data with the information that road consumptions

DISCUSSION CONTINUED ON PAGE 594

Combustion-Chamber

RESULTS of an investigation directed toward determining why deposits increase antiknock requirement are discussed here. Data are presented which indicate that substantially 100% of the increase in octane-number requirement caused by deposits results from a combination of thermal and volume effects.

An analysis procedure is given which indicates that deposit-thermal effects may result entirely from the heat that is stored in the deposits. Thus, the deposits absorb heat during the combustion process in one cycle and transfer it to the fresh charge during the intake and compression portions of the next cycle.

The findings reported in this paper show that those engines with the smallest area of combustion-chamber surface, for a given displacement, would be expected to have the smallest thermal effects and hence should have minimum deposit effects.

FUEL knock in gasoline engines has long been recognized as one of the important barriers that has limited the compression ratio at which an engine could be operated.[1, 2] Considerable progress in overcoming this barrier has been made through engine and fuel developments, as shown by the increase in average compression ratios of engines from about 4.8 to 7.5/1 during the period from 1930 to 1953.[3, 4]

An important problem yet to be solved, however, is the tendency of combustion-chamber deposits to aggravate fuel knock. Results of tests on the road[5] have shown that the presence of equilibrium deposits can be expected to increase the antiknock requirement of passenger-car engines by about 10 octane numbers. From the engine builder's viewpoint, deposits are responsible for lower compression ratios than would otherwise be used, while from the petroleum refiner's viewpoint they represent less useful octane numbers. The efficiency of engines could be improved by the benefits from compression ratios approximately 1½ units higher if the effects of combustion-chamber deposits on the antiknock requirements of engines could be eliminated. In the current octane-number range, these benefits would amount to approximately a 5% increase in the fuel-utilization efficiency of engines.

Since the effects of deposits are of such vital interest to both the engine builder and the refiner, investigations directed toward determining why deposits increase antiknock requirement have been in progress for some time at the Ethyl Corp. This paper discusses the results of some of these investigations.

Combustion-chamber deposits are also responsible for a second problem, that of deposit-induced ignition. This problem is a subject in itself, and while it cannot be separated completely from the increase in ordinary knock due to deposits, it will not be discussed in this paper.

Causes of Increase in Antiknock Requirement

A number of excellent papers presented in the last few years have offered explanations of why deposits increase antiknock requirement. The information presented in some of these papers[6, 7, 8] suggests that increases in octane-number requirement may be caused by three main deposit effects:

1. Thermal – heating of the intake charge either by heat transfer from the deposits or because of the insulating effects of the deposits.
2. Volume – increase in compression ratio caused by the physical volume of the deposits.
3. Catalysis – catalyzing the reactions leading to knock.

There is ample evidence that deposits have a thermal-insulating effect and thereby reduce the amount of heat transferred through the combus-

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[1] See *Automobile Engineer*, Vol. 9, January, 1919, pp. 2-5: "Paraffin as Fuel." by H. R. Ricardo.

[2] See *SAE Journal*, Vol. 4, April, 1919, pp. 263-269: "More Efficient Utilization of Fuel," by C. F. Kettering.

[3] See SAE *Quarterly* Transactions, Vol. 6, April, 1952, pp. 290-303: "Economic Relationship of Engine-Fuel Research," by C. L. McCuen.

[4] See "Brief Passenger-Car Data." Published by Ethyl Corp., New York, 1953.

[5] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 557-566: "Factors Affecting Octane-Number Requirement," by H. J. Gibson.

[6] See SAE Transations, Vol. 61, 1953, pp. 361-377: "Combustion-Chamber Deposition and Knock," by H. J. Gibson, C. A. Hall, and D. A. Hirschler.

[7] See SAE *Quarterly* Transactions, Vol. 5, October, 1951, pp. 565-576: "Possible Mechanisms by Which Combustion-Chamber Deposits Accumulate and Influence Knock," by L. F. Dumont.

[8] See *Industrial and Engineering Chemistry*, Vol. 43, December, 1951, pp. 2841-2844: "Gasoline Combustion," by E. C. Hughes, S. M. Darling, J. D. Bartleson, and A. R. Klingel, Jr.

Deposits and Octane-Number Requirement

Joel Warren, Ethyl Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

tion-chamber walls to the coolant. Such an insulating effect has been shown in at least one instance[7] to account for 50% of the increase in octane-number requirement caused by the accumulation of deposits. However, there is a feeling that the major portion of the thermal effects of deposits may be the result of a heat-capacity effect. Here the deposits act as a heat reservoir, absorbing heat during the combustion process in one cycle and then transferring the heat to the fresh charge during the intake and compression portions of the following cycle.

The increase in octane-number requirement, caused by the change in compression ratio which results when deposits are accumulated, is easily obtained by measurement of the octane-number requirement at the equilibrium-deposit condition, and again after the compression ratio has been lowered to the clean-engine value. As much as 10 to 40% of the requirement increase which resulted from deposits has been reported[7] to be caused by volume effects.

In most reported instances, the sum of the thermal and volume effects has been found to be responsible for the major portion of the increase in octane-number requirement. Catalytic effects, if they existed, were believed to account for only a very small portion of the increase in octane-number requirement.

Although the recent papers referred to offer explanations as to why deposits increase antiknock requirement, it was believed that additional evidence would be desirable. The single-cylinder engine test results discussed in this paper add considerable information about the mechanism by which deposits influence knock. Special emphasis is given to the question of whether or not the thermal effects of deposits are the result of a thermal-insulating effect or of a heat-capacity effect.

Equipment and Test Procedure

Heat-Transfer Measurements – The engine used for all of the tests discussed in this paper was a single-cylinder, CFR knock-test engine with the standard overhead-valve cylinder replaced by an L-head cylinder. In order to obtain data on heat transfer during deposit accumulation, the engine was modified so that the heat lost to the coolant in the cylinder head and that lost to the coolant in the engine block could be measured separately during operation. In order that the heat transferred to these regions could be measured separately, a Lauson-engine coolant condenser was piped to the block, while a CFR-engine coolant condenser was connected to the cylinder head.

Fig. 1 shows the general location of these units on the engine. Solid brass gaskets, 0.003 in. thick, were used above and below the regular head gasket to separate the coolant in the block from that in the head. This provided separate evaporative-type

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Table 1 – Operating Conditions Used in Single-Cylinder L-Head Engine Cycling Tests

	Load	
	Idle	Full Throttle
Cycle Duration, .e	50	150
Speed, rpm	600	900
Fuel/Air Ratio	0.087	0.077
Ignition Timing	Tdc	
Jacket Coolant Temperature, F	150	
Oil Temperature, F	160	
Intake Air:		
Temperature, F	112	
Moisture, grains per lb	34	
Pressure	Atmospheric	
Compression Ratio (Clean)	7.0	
Test Duration, hr	165[a]	

[a] Equilibrium octane-number requirement was usually reached before 100 test hr. Determinations of octane-number requirement were made at the full-throttle engine conditions.

cooling systems for both the head and the block. The block and both condensers were insulated to reduce the effects of changes in room temperature on the heat-transfer values. In order to allow comparisons with other data taken previously, methanol was used as the coolant for each of the two systems, except in a few special cases in which coolant mixtures of various boiling points were used.

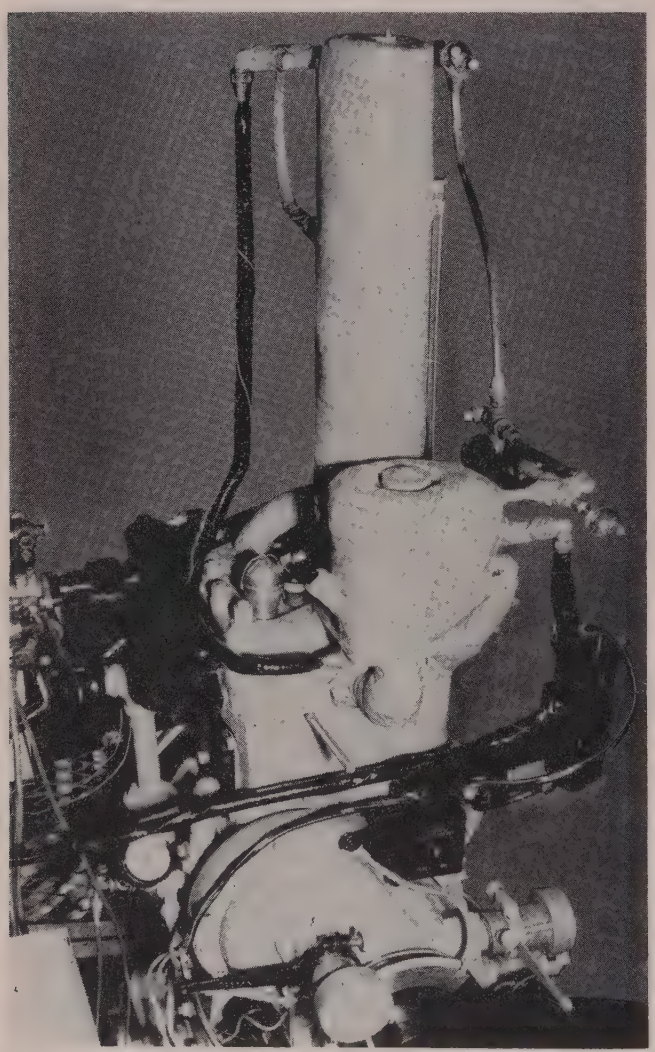

Fig. 1 – Modified CFR test engine showing special arrangement of cooling system

Isooctane was used as the fuel during all heat-transfer measurements. Engine operating conditions were those shown for the full-throttle portion of Table 1. The actual measurements of the heat transferred to the coolants were made by determining the amount of heat lost to the cooling water used to condense the coolant in each of the evaporative-type systems. Check runs showed good reproducibility with this equipment. Fig. 2 is a schematic representation of the systems just discussed.

Compression-Ratio Measurements and Effects – The increase in compression ratio that takes place as combustion-chamber deposits are accumulated influences the antiknock requirement of the engine. In order to determine this effect, the engine was rated at the equilibrium-deposit condition and then again with the deposits still in place but with the compression ratio lowered to the clean-engine value. The clean-engine compression ratio with deposits present was obtained by adding the necessary thickness of gaskets between the engine block and the crankcase. A sonic device, by means of which the clearance volume could be measured without disturbing the cylinder, was used to measure this volume both with and without deposits present. The condition of the deposits was not disturbed during the adjustment of the compression ratio when this procedure was used.

The difference between the antiknock requirements at the two conditions was considered to be caused by the change in compression ratio that took place when the deposits were accumulated, and it is referred to as deposit-volume effect. Deposit-volume effects determined in this manner were measured under conditions where the octane number of the rating fuels was either at or close to the equilibrium octane-number-requirement level.

Deposit-Accumulation Schedule – Since the primary objective of the work reported in this paper was concerned with the effects of combustion-chamber deposits, a single-cylinder engine deposit-accumulation schedule was desired which would give both reproducible test results and which would rate oil-, fuel-, and tel-deposit effects in the same relative manner as passenger cars in light-duty road service. A previous paper[6] has reported such an engine operating procedure. This procedure, the details of which are included in Table 1, was used during the deposit-accumulation phase of all but one test reported in this paper. The retarded ignition timing tends to increase both the amount of deposits formed and the effects of the deposits on knock. As a result, an expanded scale is obtained which makes it easier to observe small effects.

Air-Flow Measurements – Air-flow measurements were accomplished by means of a precision-type

variable-area flowmeter, which was installed ahead of a control valve in the air line leading to the carburetor. Air pressure at the flowmeter was maintained at 45 psi gage, and the temperature at 125 F. After passing through the meter, the air was expanded through the control valve to atmospheric pressure in a combination surge and heater tank, which was connected to the air horn of the carburetor. Since the air pressure in the combination surge and heater tank was always below the critical pressure passed through during the expansion process, pulsations caused by the single-cylinder engine operation did not influence the functioning of the flowmeter. Hence, since steady-flow conditions existed at the meter, the airflow measurements are believed to have been highly accurate.

Preliminary Investigations

Typical test results using the single-cylinder CFR L-head engine and operating schedule referred to in Table 1 are shown in Figs. 3 and 4. A nonleaded commercial-type automotive fuel and a nonadditive high-V.I. oil were used as the fuel-oil combination in this test. Antiknock requirement, airflow, compression ratio, and the amount of heat transferred to the engine coolants, all changed during the initial stages of the deposit-accumulation period and finally reached an equilibrium value near the 100-hr point.

Fig. 5 includes antiknock requirements determined at four different ignition timings when the engine contained these equilibrium deposits. Shown also in Fig. 5 are antiknock requirements at the same ignition timings when the engine contained the equilibrium deposits and the compression ratio was lowered to the start-of-test clean-engine value. The difference between the two requirements obtained at a given ignition timing was considered to be caused by deposit-volume effects.

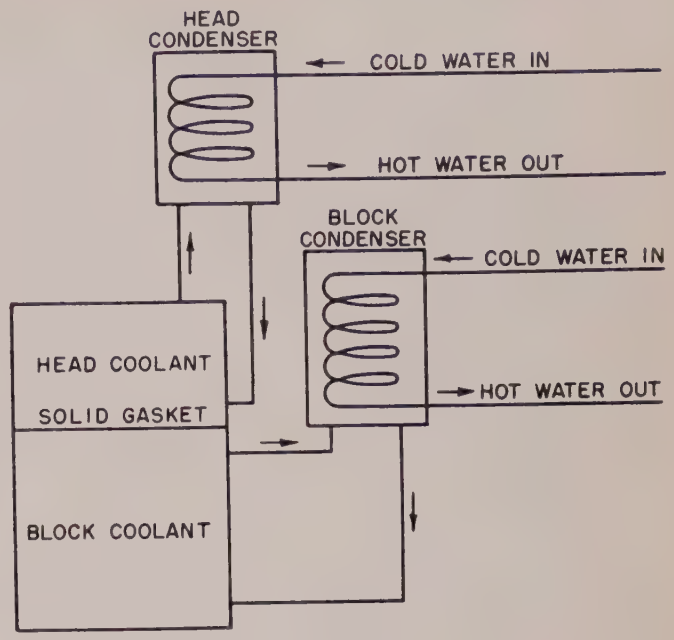

Fig. 2 – Arrangement of equipment used to measure heat transfer to coolant

It is significant to note from Fig. 3 that the air consumed by the engine decreased as deposits were accumulated and that it reached a stabilized condition at about the same time that the equilibrium octane-number requirement was reached. After the combustion-chamber deposits had been removed at the end of the test, the airflow was substantially the same as at the clean-engine condition at the beginning of the test. This uniformity of airflow indicated that any induction-system

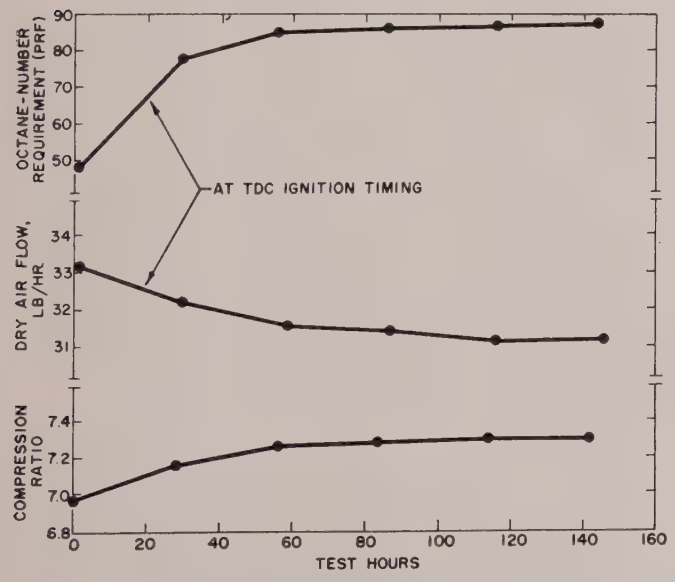

Fig. 3 – Effect of deposit accumulation on certain engine variables

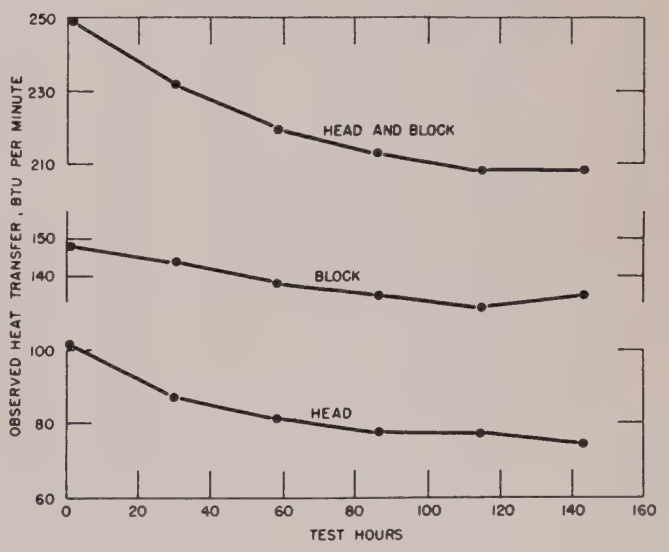

Fig. 4 – Effect of deposit accumulation on observed heat transfer to coolant

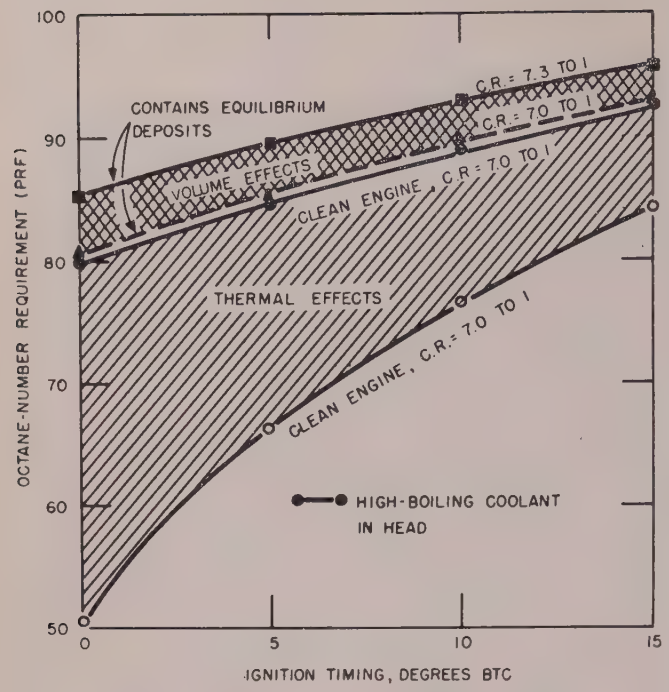
OCTANE-NUMBER REQUIREMENT (PRF)
IGNITION TIMING, DEGREES BTC
CONTAINS EQUILIBRIUM DEPOSITS
VOLUME EFFECTS
THERMAL EFFECTS
C.R. = 7.3 TO 1
C.R. = 7.0 TO 1
CLEAN ENGINE, C.R. = 7.0 TO 1
CLEAN ENGINE, C.R. = 7.0 TO 1
HIGH-BOILING COOLANT IN HEAD

Fig. 5 – Influence of ignition timing on thermal and volume effects of deposits

deposits accumulated during the test were not sufficient to affect the volumetric efficiency of the engine. It seemed reasonable to conclude, therefore, that the entire reduction of airflow when the engine contained deposits was caused by a heating of the incoming charge by the deposits.

It was believed that this heating effect of the deposits could be simulated in a clean engine by increasing the jacket-coolant temperature and thereby raising the temperature of the surface of the combustion chamber. The action of the hotter surface of the combustion chamber should be to decrease the volumetric efficiency by heating the incoming charge in a manner similar to that which takes place when the engine contains deposits. Such an effect should influence the octane-number requirement of the engine, possibly to the same extent as did the presence of deposits.

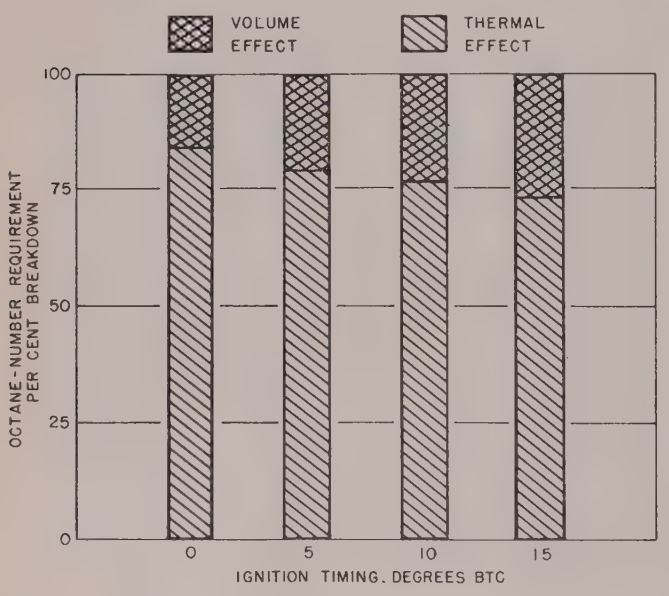

Fig. 6 – Contribution of thermal and volume effects to total octane-number-requirement increase

In order to check this hypothesis, an investigation was made in which the clean engine was run with a high-boiling coolant in the head. Methanol was used as the coolant in the block, in order that piston and piston-ring clearances and other related effects would be similar to those in the other phases of the test program while deposits were being accumulated. The influence of the hot combustion-chamber surface in the head was just enough to cause an airflow which was the same, within experimental error, as that which resulted when the compression ratio of the engine with deposits was adjusted to the start-of-test clean-engine value.

Fig. 5 shows the antiknock requirements of the clean engine determined at four different ignition timings when the high-boiling coolant was used in the head. These antiknock requirements were substantially the same as those which occurred when the engine contained deposits and the compression ratio was lowered to the clean-engine value. Since the airflow had been reduced to the same level, in one case by the hot deposits and in the other by the hot surface of the combustion chamber, the resulting increases in octane-number requirement above the clean-engine, normal-coolant condition were believed to have been caused by thermal effects. The magnitudes of these thermal effects are shown in Fig. 5. Thermal effects are discussed in greater detail in a later section of this paper under the heading "Deposit Effects." Data are presented there which indicate that deposit-thermal effects are undoubtedly the result of heat-capacity effects. It is obvious that when the limits of experimental error are considered, the sum of the thermal and volume effects has been shown to account for substantially 100% of the increase in octane-number requirement that was caused by the accumulation of the deposits, as shown in Fig. 5.

Fig. 6 shows the same data as Fig. 5, but here the volume and thermal effects are presented as percentages of the total increase in octane-number requirement at a given ignition timing. Viewed in this manner, volume effects are observed to increase and thermal effects to decrease as ignition timing is advanced. The maximum-power ignition timing for this engine was about 17 deg btc when the engine contained the equilibrium deposits.

Engine Air Consumption and Antiknock Requirement

The octane-number requirement of an engine operated under a given set of conditions is influenced by the end-gas temperature of the working medium. The test results just discussed suggest

that the end-gas temperature of the charge was the same when the engine contained the equilibrium deposits at the clean-engine compression ratio condition as when the deposit condition was simulated by operating the engine at an elevated head-coolant temperature.

The accumulation of deposits in an engine, exclusive of volume effects, may cause an increase in the end-gas temperature of the charge and hence in the octane-number requirement by one or more of the following methods:

1. The deposits may absorb heat from one cycle and transfer it to the incoming charge during the suction stroke of the following cycle.
2. The deposits may also heat the charge during the compression stroke.
3. The deposits may act as thermal insulation for part of the compression stroke and during combustion and thereby reduce the amount of heat dissipated from the charge to the coolant.

The action of the hot surfaces of the combustion chamber brought about in the clean engine by elevating the head-coolant temperature could also affect end-gas temperature and hence octane-number requirement by one or more of these three methods. In method 3, the reduction in heat dissipated would be the result of the increased coolant temperature and not the result of an insulating effect. However, future reference to this action will be referred to as an insulating effect.

The transfer of heat from the deposits to the incoming charge during the suction stroke (method 1) would result in a decrease in the density of the charge. This effect would be recognized as a reduction in the breathing capacity or volumetric efficiency of the engine. However, heating of the charge by the deposits during the compression stroke, and any thermal-insulating effect of the deposits during the compression stroke and the combustion process, would not be expected to influence directly the volumetric efficiency of the engine, since they both would occur after the intake valve had closed. Such effects might, however, have a small influence on airflow as a result of changes in clearance-gas temperature.

In order to furnish background information concerning these possible effects, a more complete investigation was made of the influence of certain engine operating conditions on the air consumption and octane-number requirement of the clean engine. Decreases in airflow to the engine were accomplished by each of the following methods:

1. By throttling
2. By raising the temperature of the carburetor air ahead of the carburetor
3. By raising the head-coolant temperature
4. By suspending stainless-steel washers inside the combustion chamber.

In all other respects, engine operating conditions for this investigation were the same as those listed for the full-throttle portion of Table 1.

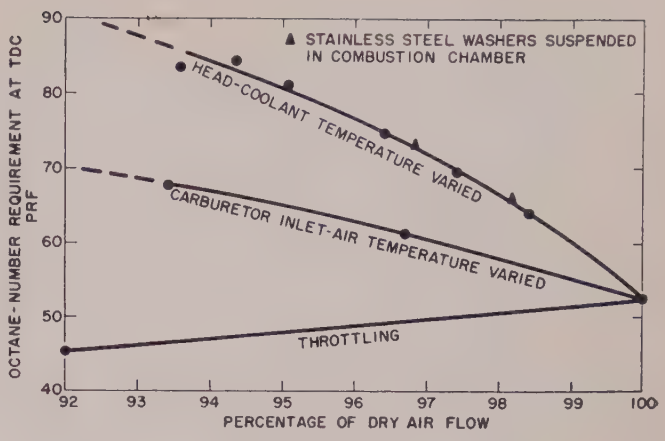

Fig. 7 – Influence of certain engine operating conditions on air consumption and octane-number requirement

Fig. 7 shows the relation between the percentage reduction in airflow and the octane-number requirement of the engine as the airflow was varied by each of the above methods. The 100% airflow condition was selected as that which occurred in the clean engine at the full-throttle conditions listed in Table 1.

Throttling was observed to decrease slightly the octane-number requirement of the engine in the range of throttling that was investigated. This effect was as expected, since if considered as a free-expansion process, throttling would be expected to reduce the charge temperature by a relatively small amount for the degree of throttling used.

Raising the temperature of the carburetor air ahead of the carburetor would reduce the density of the charge by a temperature effect and result in a lower volumetric efficiency. However, when compared to the throttling method at the same rate of airflow, the temperature of the charge at the end of the suction stroke would be increased; hence, the end-gas temperature of the charge would be greater. As a result, the octane-number requirement of the engine would be expected to be higher. The extent to which this effect occurred

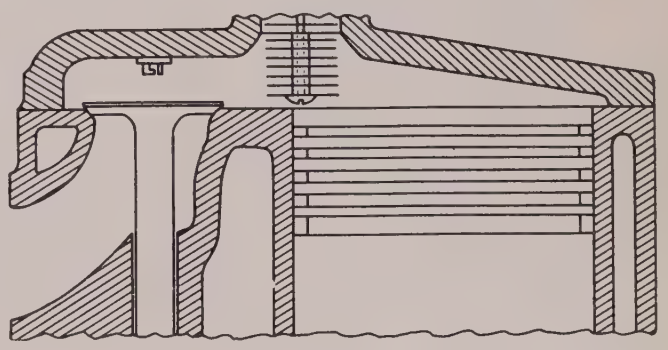

Fig. 8 – Arrangement of steel washers in combustion chamber of CFR L-head engine

in the clean engine is shown in Fig. 7.

Raising the head-coolant temperature or suspending steel washers in the combustion chamber as shown in Fig. 8 and assuming the volume of the washers to be negligible should give the same charge temperature at the end of the suction stroke for a given reduction in airflow as was obtained by raising the carburetor air temperature. However, it would be expected that the charge temperature would be increased further, during at least a portion of the compression stroke, by the action of the hotter combustion-chamber surface in one method and by the hot washers in the other. The resulting end-gas temperatures would be increased as a result of these effects. That this undoubtedly took place is indicated from the results, plotted in Fig. 7, which show that both of these methods gave the greatest increase in octane-number requirement of the clean engine at a given reduction in airflow.

When steel washers were used, the entire heating of the charge was due to a direct heat-transfer process. This did not involve a thermal-insulating effect, since the presence of the steel washers had no measurable effect on the heat transferred to the coolant. An insulating effect was possible when the head-coolant temperature was elevated, because the rate of heat transfer to the coolant was automatically decreased as the coolant temperature was increased. Since a given reduction in airflow by either of the two methods produced the same octane-number requirement, it is possible that the increased thermal-insulating effect of the combustion-chamber walls when the head-coolant temperature was elevated did not influence the octane-number requirement of the engine. Hence, the entire increase in octane-number requirement as the air consumption was reduced can be considered to have been the result of the ability of the steel washers in one method and the hot surface of the combustion chamber in the other to store up heat from one cycle and transfer it to the incoming charge on the following cycle. This effect has been defined as a heat-capacity effect. The assumed action of the surface of the combustion chamber is supported by the work of Eichelberg,[9] who has reported that the temperature of the inner surface of a certain combustion chamber fluctuated by as much as 50 F during one complete engine cycle.

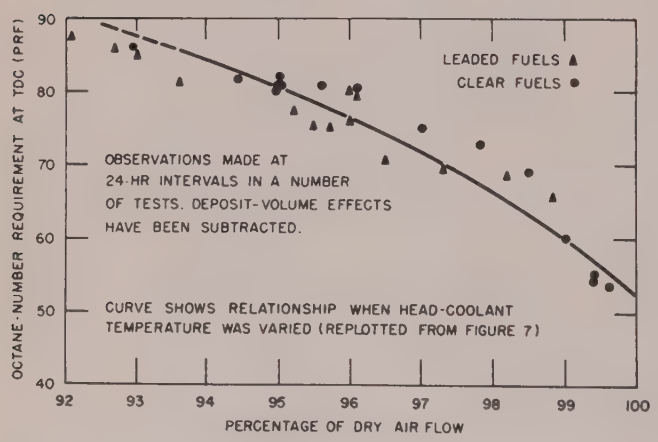

Fig. 9 – Influence of engine deposits on air consumption and octane-number requirement

Deposit Effects

The work reported in the previous section of this paper, combined with the discussion concerning the data in Fig. 5 suggests that deposit-thermal effects may be the result of the heat-capacity of the deposits, whereby the incoming charge is heated by the energy that was stored up in the deposits from the previous cycle. If this is true, then the relation between octane-number requirement and reduction in airflow shown in Fig. 7, for the method where the head-coolant temperature was elevated, should also apply in a given test, after deposit-volume effects are considered, to all portions of the deposit-accumulation period from the clean-engine condition to that including the equilibrium-deposit condition.

Fig. 9 shows the octane-number requirement, after volume effects are subtracted, for a number of deposit-accumulation tests, superimposed upon the portion of Fig. 7 in which the head-coolant temperature was elevated. Various combinations of oil with leaded and nonleaded fuels were used in these tests. The deposit-accumulation schedule shown in Table 1 was used in all but the one test in which the engine was cycled under heavy-duty operating conditions.

Measurements of airflow, octane-number requirement, compression ratio, and heat transferred to the coolant were made in each test at about 24-hr intervals. The volume effect at the end of each of these intervals in a given test was calculated by multiplying the end-of-test volume effect by the fractional increase in compression ratio that existed at the end of the particular test period. With few exceptions, the deposit data in Fig. 9 are in good agreement with those data obtained when the head-coolant temperature of the clean engine was elevated.

These results indicate that the deposit-thermal effects in the tests reported were primarily the result of the heat capacity of the deposits. It was recognized, however, that the heat-capacity effects of deposits might be dependent on the magnitude of the thermal-insulating effect of the deposits. Those deposits with good thermal-insulating properties might be expected to have high surface temperatures and hence great heat-capacity effects. If this relation were true in all instances, the

[9] See *Engineering*, Vol. 148, Oct. 27, 1939, pp. 463-466: "Some New Investigations on Old Combustion-Engine Problems," by G. Eichelberg.

deposit-requirement data shown in Fig. 9 should fall on one representative curve if plotted against heat transferred to the coolant. Fig. 10 shows that this did not happen. The limits of experimental error are not great enough to account for the great spread of data in Fig. 10.

These results indicate that it might be possible for certain deposits to have large insulating properties and yet cause a small increase in octane-number requirement. The reverse of this would also be a possibility. Deposits which are compact and evenly distributed in the combustion chamber might have comparable thermal-insulating and heat-capacity effects. However, the "corn-flake" type of deposit might be expected under certain conditions to have relatively poor thermal-insulating properties but large heat-capacity effects. The raised edges of the corn flakes may expose fairly large masses of the deposits to the hot gases during combustion and at the same time reduce the cross-sectional areas of the heat paths. In addition, the surfaces of the combustion chamber exposed by the curled edges of the corn flakes, as well as the exposed surfaces of the head when actual flaking occurs, leave a better heat path to the coolant. Thus, in an extreme case, the corn-flake type of deposit might result in a large heat-capacity effect and at the same time have relatively poor thermal-insulating properties.

Other factors that might influence the relationship between thermal-insulating and heat-capacity effects include:

1. The distribution or location of the deposits in the combustion chamber.
2. The composition of the deposits.

Test results plotted in Fig. 11 show a case in which changing to a different fuel, after deposit equilibrium was reached, resulted in a relatively small decrease in octane-number requirement but a large change in heat transferred to the coolant. In this case, the initial operation of the engine was on clear toluene with a high-V.I. nonadditive oil as the crankcase lubricant. The engine was operated on the schedule listed in Table 1 until deposit equilibrium was reached. The octane-number requirement increased by about 40 units, and the heat transfer to the head coolant decreased by about 24% when compared to the clean-engine values. The fuel was then changed to clear isooctane, and operation of the engine continued until a new equilibrium condition was established. The new octane-number requirement was only 7 octane units less than that with toluene, but the heat-transfer rate to the head coolant was restored to very nearly the clean-engine value. No appreciable change occurred in the heat transferred to the block coolant.

This is a case in which the resulting deposits had a very small thermal-insulating effect but caused a large octane-number requirement. Airflow to the engine varied in a manner which indicated

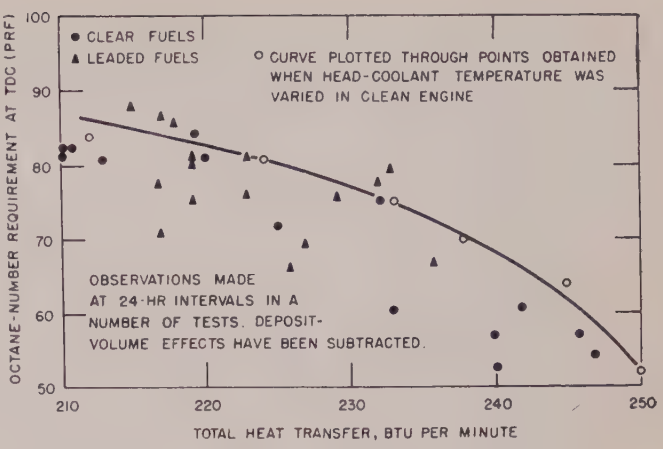

Fig. 10 – Relation between heat transfer to coolant and octane-number requirement

that the resulting requirement, after volume effects had been considered, was due to heat-capacity effects. These results show that it is possible for deposits to have a large heat-capacity effect, with resulting high octane-number requirement, and at the same time produce relatively small thermal-insulating effects.

The results of the tests just discussed indicate that:

1. A correlation exists between the decrease in airflow caused by the presence of deposits and the octane-number requirement of the engine after volume effects are subtracted.
2. A correlation does not exist in deposit-accumulation tests in the given engine between octane-number requirement and heat transferred to the coolant after volume effects are considered.
3. Deposit-thermal effects were undoubtedly the

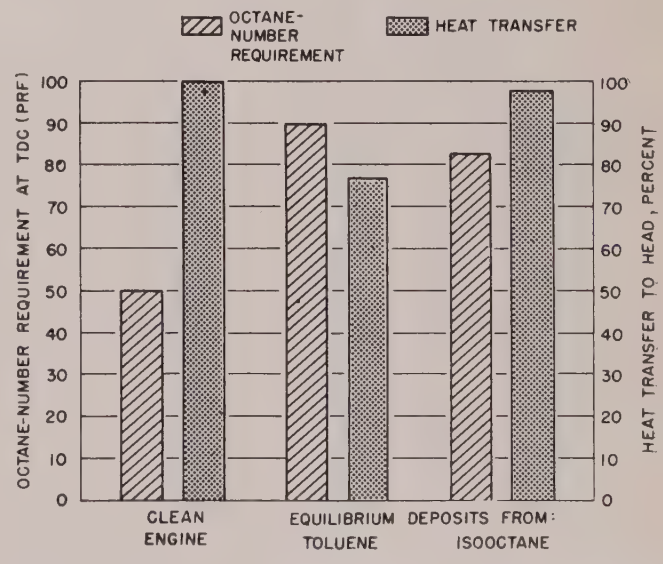

Fig. 11 – Thermal-insulating effects of certain deposits and octane-number requirement. Deposits from toluene left in engine when fuel was changed to isooctane

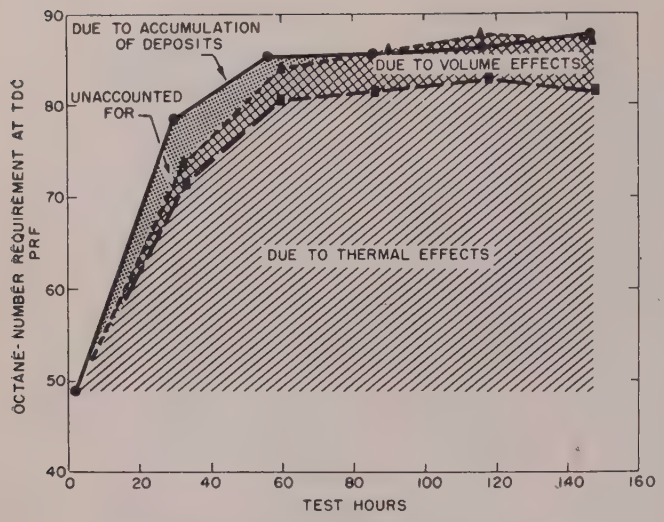

Fig. 12 – Contribution of thermal and volume effects of deposits to total increase in octane-number requirement

result of a heat-capacity effect; that is, the ability of the combustion-chamber deposits to store up heat from one cycle and transfer it to the incoming charge on the following cycle. It is possible that catalysis may have had a bearing on the magnitude of the heat-capacity effects.

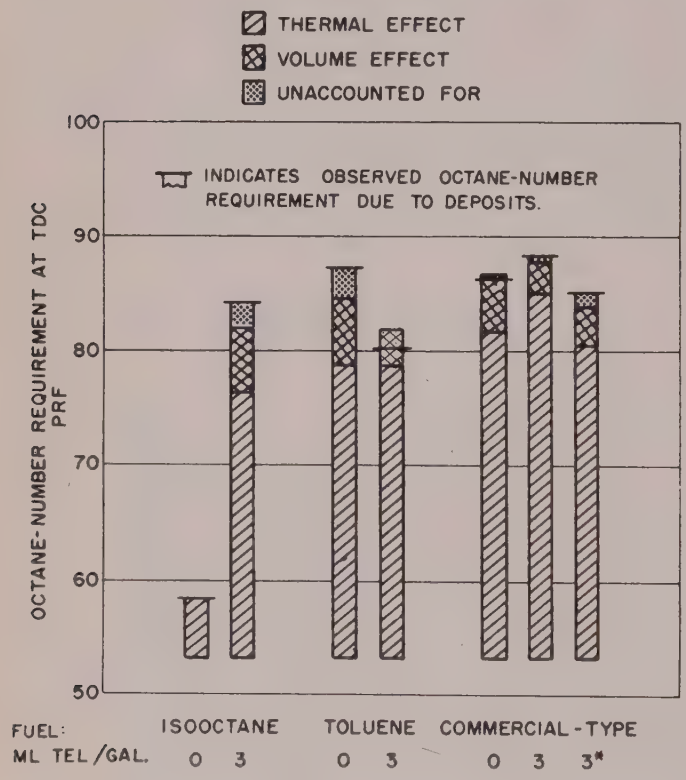

Fig. 13 – Contribution of thermal and volume effects of deposits to total increase in octane-number requirement for various fuels. Bar at extreme right plotted from data obtained under heavy-duty operating conditions

4. The relation shown to exist in Fig. 7, between airflow and octane-number requirement when the head-coolant temperature of the clean engine was elevated, can be used to evaluate that portion of the increase in octane-number requirement that is caused by a thermal effect during any period of a deposit-accumulation test in the given engine.

Fig. 12 shows typical results obtained when the relation listed in item 4 was used to evaluate that portion of the increase in octane-number requirement that was caused by thermal effects in a particular deposit-accumulation test. The test was run in the CFR L-head engine using the deposit-accumulation schedule listed in Table 1. A nonleaded commercial-type fuel and a high-V.I. nonadditive oil were used as the fuel-oil combination. Measurements of airflow, compression ratio, heat transferred to the coolants, and octane-number requirement were made at approximately 24-hr intervals. After the completion of the test, calculations were made of the percentages of the clean-engine airflow that existed at the end of each of the 24-hr intervals. These dry-airflow percentages were corrected for the effect of any induction-system deposits. By using the upper curve in Fig. 7, dry-airflow percentages were used to determine the octane-number requirement due to the thermal effects of the deposits.

The results in Fig. 12 show that these thermal effects accounted for a large portion of the increase in octane-number requirement caused by the accumulation of the deposits. The deposit-volume effect at the end of each 24-hr interval was calculated by multiplying the deposit-volume effect at the end of the test by the fractional increase in compression ratio that existed at the end of the particular test period. The results in Fig. 12 show that the sum of the thermal and volume effects, as defined, accounted for most of the increase in octane-number requirement that resulted from the accumulation of the deposits. Small errors in airflow measurements could cause the amount of unaccounted-for octane-number requirement that existed. The same general results occurred when a similar procedure was applied to the requirement data taken at other ignition timings.

Since deposits generally are of major concern only after equilibrium has been established, increases in octane-number requirement, deposit-volume effects, and deposit-thermal effects have been averaged over the last half of the deposit-accumulation period and are shown as a bar chart in Fig. 13. Table 2 shows these same volume and thermal effects as a percentage of the increase in total octane-number requirement.

Shown also in Fig. 13 and in Table 2 are results obtained by using the same procedure for averaging the results of tests run on other fuels but with the same type oil. All test data have been corrected to a common start-of-test clean-engine octane-number requirement.

Clear isooctane was run under low oil-consumption conditions and produced no measurable deposit-volume effect and very little increase in octane-number requirement. Thermal effects accounted for 100% of the increase in octane-number requirement. However, leaded isooctane produced considerable deposits and had the largest volume effect in all tests reported. This was true both in terms of increase in octane units and in percentage of total increase in octane-number requirement.

Toluene gave the most erratic results. The erratic results may have been due to the fact that operation of the engine for only short periods of time on primary reference fuel blends, during the rating or heat-transfer measurement periods, disturbed the condition of the deposits to a greater degree than with any other fuel tested. This was especially true with clear toluene. When the engine had been run previously on clear toluene, operation on isooctane for only a short period of time reduced the octane-number requirement by as much as 10 octane numbers.

The results obtained with the commercial-type fuel show no great difference in thermal and volume effects, whether the fuel was leaded or nonleaded. Virtually similar thermal and volume effects were obtained when the engine was operated under heavy-duty conditions.

Deposit-volume effects accounted for from 0 to 30% of the increase in octane-number requirement, depending upon the fuel used and the engine operating conditions. Deposit-thermal effects accounted for from 70 to 100% of the increase in octane-number requirement, depending upon the same variables. The commercial-type fuel gave results which were approximately in the middle of the range of the values shown for the thermal and volume effects.

A previous paper[6] presented data which indicated generally that those engines with a minimum ratio of combustion-chamber-surface area to piston displacement can be expected to have a minimum increase in octane-number requirement from the accumulation of combustion-chamber deposits. Based upon the findings reported in this paper, those engines with the smallest area of combustion-chamber surface, for a given displacement, would be expected to have the smallest thermal effects and hence, as reported, should have minimum deposit effects.

Table 2 – Contribution of Thermal and Volume Effects to Total Increase in Deposit Octane-Number Requirement

	Percentage Increase in Octane-Number Requirement[a]					
	Clear Fuel			3.0 ml TEL per Gallon		
	Volume Effect	Thermal Effect	Unaccounted For	Volume Effect	Thermal Effect	Unaccounted For
Isooctane	0	100	0	19	74	7
Toluene	17	75	8	11	95	−6
Commercial-Type Fuel	14	87	−1	8	92	0
Commercial-Type Fuel[b]	—	—	—	11	85	4
Average	10.3	87.4	2.3	12.2	86.5	1.3

[a] CFR L-head engine. Octane-number requirement data obtained at tdc ignition timing. Deposit-accumulation schedule is as listed in Table 1.
[b] Deposits accumulated using heavy-duty engine operating conditions.

Conclusions

1. The sum of the thermal and the volume effects has been shown to account for substantially 100% of the increase in octane-number requirement that was caused by the accumulation of deposits in a number of different tests.

2. Deposit-thermal effects have been shown to be the result of a heat-capacity effect, wherein the incoming charge is heated by energy that was stored in the deposits from the previous cycle.

3. Deposit-volume effects were obtained by determining the difference between the octane-number requirement at the deposit-equilibrium condition and after the compression ratio had been adjusted to the clean-engine value.

4. The thermal effects of deposits accounted for from 70 to 100% of the increase in octane-number requirement, depending upon the type of fuel used and the engine-operating conditions.

5. The volume effects of deposits accounted for from 0 to 30% of the increase in octane-number requirement, depending upon the same variables as in item 4.

6. Tests run with a leaded and a nonleaded commercial-type fuel gave deposit-thermal and deposit-volume effects that were approximately in the middle of the range of values indicated in items 4 and 5.

7. The relation between airflow and octane-number requirement was shown to be the same when the head-coolant temperature of the clean engine was elevated as when deposits from various fuels were accumulated in the engine, provided that deposit-volume effects were subtracted.

8. While it is generally believed that deposits with good thermal-insulating properties should be expected to have large heat-capacity effects, in at least one instance, certain deposits which had a very small insulating effect produced a large heat-capacity effect. The location of the deposits in the combustion chamber, the type of structure of the deposits, and their composition may have been influencing factors in this case.

9. Based upon the findings reported in this paper, those engines with the smallest area of exposed combustion-chamber surface, for a given displacement, would be expected to have the smallest thermal effects, and hence, should have minimum deposit effects.

DISCUSSION ON FOLLOWING PAGE ——→

DISCUSSION

Discusses Catalytic Effects Of Deposits on Combustion

– E. C. Hughes
Standard Oil Co.

THE author is to be congratulated for an unusual experimental approach to one of the fundamental problems of internal-combustion engines. This work is marked by the clear-cut logic of its deductions. We are all vitally interested in "how so little can do so much to so many." Thus, 0.04 lb of a thin layer in a 500-lb engine raises the cost of gasoline manufacture by about $300 million a year. In fact, we could get along without any tel in our industry if the harm caused by deposits could be eliminated.

For several years researches have been carried on to find the main reason they cause the trouble. Like many phenomena in nature we are now becoming aware that there is no one cause of the harm of these deposits. The increase in compression ratio which these deposits cause was found to be only a minor part of the cause of the trouble.[a] Further and very elegant proof for this is also presented by Mr. Warren.

Beyond this, Mr. Warren postulates that the deposits hold and transfer heat to the fresh charge and that is the major cause of deposit harm. While part of the effect of deposits might be a simple provision of heat to the gases, it would seem highly possible that another part would be due to a catalytic effect of the deposits on the heat-releasing precombustion reactions. Thus Cornelius and Caplan[b] calculated a 75-deg increase in unburned-gas temperature in the presence of deposits. They also note an increase in rate of combustion in the presence of deposits (maximum power is obtained at a more retarded ignition setting than when deposits are absent). When the charge is heated by the deposits (or by washers) it most certainly will undergo more extensive precombustion reactions. These would add appreciably to the temperature and content of knock-producing materials in the end gas.

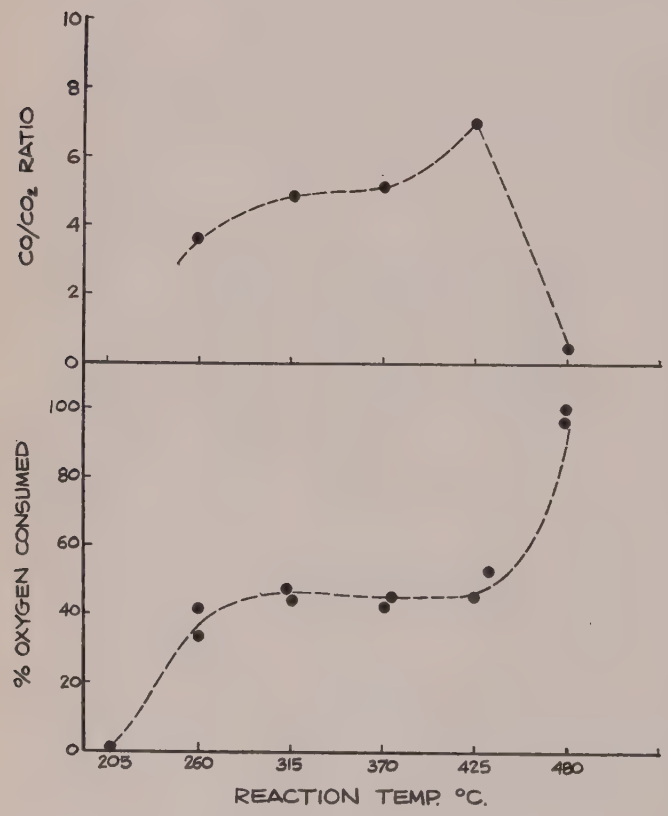

Fig. A – Oxidation of n-heptane in open tube

Fig. A shows the effect of temperature on the oxidation of heptane in an open tube which we gave in a paper before the American Chemical Society in 1951.[c] The left-hand side of the lower curve up to 316 C shows a rapid increase in rate so that a difference of only 15 C results in the doubling of the amount oxidized. If the engine conditions are such that the end-gas temperature is in this region, then seemingly minor changes in those conditions will cause rapid changes in such things as knock rating.

Above 316 and on up to about 430 C, change in conditions will have a lesser effect because the reactions tend to go only so far and then stop by some self-inhibitor mechanism. However, above 430 C the rate of reaction is very temperature sensitive. Now we have found that in this temperature region various solids have powerful catalytic actions. Some of these are commonly found in engines. Thus several of them, PbO, $PbCl_2$, B_2O_3, SiO_2, were shown to have a powerful inhibiting effect on the oxidation of heptane in a tube at a temperature of 900 F. One of them, Fe_2O_3, did not. Others that have been noted to have an inhibiting effect were $PbSO_4$, KCl-coated steel, aluminum, chrome plate, lead-borate-coated objects.

Carbon and carbon containing lead deposits from engines could not be run because the carbon itself burned. Extrapolation of results on various carbons at lower temperature indicated that it was only slightly inhibitory of the oxidation of heptane.

Dumont[a] found many of these materials also to show a catalytic (antiknock) effect when fed as dusts to an engine. Our work suggests that he was utilizing these surfaces at a higher temperature than they exist on the walls of an engine. While our paper showed the "negative" catalytic effects of these solids at 900 F, it also showed but small differences between them at 600 F or below. Except for the high points the composition of deposits[a, d] has been found to be such that they must be below 600 F. Otherwise the carbon would be absent, particularly in the presence of Pb residues. [e, f]

At such temperature the major difference noted in our paper was that a low surface (open glass tube) or extended surfaces of SiO_2, $PbSO_4$, and B_2O_3 did not convert CO to CO_2. On the other hand, many of the materials in engine deposits and the deposits themselves were powerful for converting CO into CO_2 at 600 F. Iron oxide or steel, carbon, PbO, $PbCl_2$, $PbBr_2$, Al, and ZnO have been among those doing this.

Thus catalytic action of one kind or another, depending on the temperature, has been observed for the solids in question. We have not tried stainless steel out of which

[a] See footnote 7 of main paper.
[b] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 488-510: "Some Effects of Fuel Structure, Tetraethyl Lead, and Engine Deposits on Precombustion Reactions in a Firing Engine," by W. Cornelius and J. D. Caplan.
[c] See footnote 8 of main paper.
[d] See footnote 6 of main paper.
[e] See *Industrial and Engineering Chemistry*, Vol. 45, July, 1953, pp. 1501-1508: "Combustion-Chamber Deposits," by J. D. Bartleson and E. C. Hughes.
[f] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-752: "Flame Photographs of Autoignition induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.

Mr. Warren prepared his washers, but steel itself is potent. Perhaps even in that experiment some catalytic enhancement of the precombustion reaction occurred. It would be of interest to try "washer" material which is not catalytic for CO to CO_2, such as a coating of $PbSO_4$ or B_2O_3 on the washers. Such an experiment might serve to sort out catalytic and heat-capacity effects.

The inclusion of catalytic effects has the advantage that it might help provide an explanation for some remaining puzzling (to me) effects. Thus, for example, the powerful effect of smaller weights of nonleaded deposits is not well explained by the work so far. Gibson et al[d], Dumont[a], and this paper have examples of this. If Mr. Warren got the same deposits reported by Gibson in the work of his Fig. 13, 3 g of carbon from toluene were as bad on ORI (octane-requirement-increase) as 21 g of leaded deposit from leaded isooctane. The toluene carbon could hold about 0.433 cal per F whereas the isooctane deposit would hold 1.13, or 2.6 times as much. Nevertheless the effects on ORI were the same. Likewise we have seen this effect in studying various fuel-oil combinations. Another troublesome fact (for the heat-capacity theory) is that removal of less than half the deposits from the end-gas area will remove much more than half the octane-requirement increase. Can heat capacity, thermal insulation, or catalytic effect better explain such phenomena?

In tube-oxidation work and from time to time in engine work, definite catalytic effects have been observed. Perhaps they are most marked only under certain conditions of deposit and end-gas temperature. Recent studies we have made on boron compounds and other "catalytic gasoline additives" have been successful in lowering octane requirement but a small part of the time. Strangely they seem much more consistent in reducing the amount of deposit. Nevertheless these observations do not rule out the possibility that some of Mr. Warren's octane requirement is contributed through catalytic action as well as heat content of the deposits.

In summary, it would seem that Mr. Warren is to be complimented for extending our horizons in so difficult and important a field. I can only hope, that as we experiment further, we all will not lose sight of the probable complex nature of phenomena and thus try to pin too much on any one mechanism.

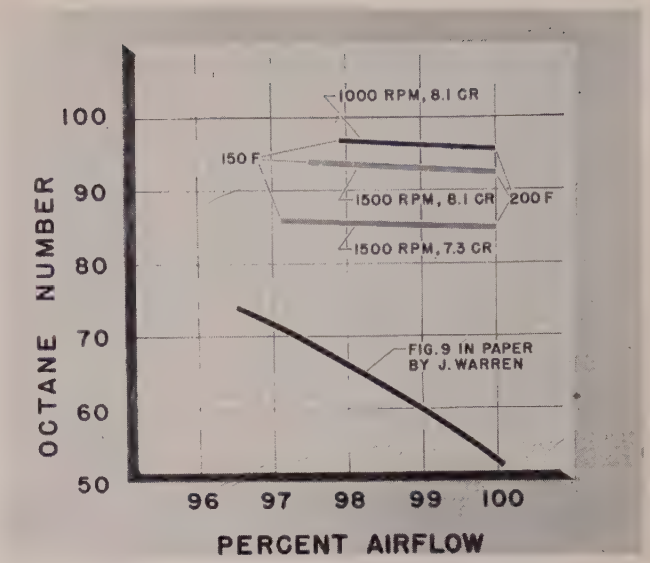

Fig. B – Octane requirement at higher coolant temperatures, no deposits, L-head engine

Lists Effects of Deposits On Power at Retarded Spark

– Otto Enoch
Ford Motor Co.

THIS paper, we feel, is a very valuable contribution to the deposit problem. In particular, the author's experimental approach to the important question whether heat capacity or thermal insulation governs the change in octane requirement after deposit buildup seems to us a very fine piece of research work.

The volumetric effects of deposits given by the author generally check with our test data. We find them varying greatly with operating conditions. For instance, in a multicylinder L-head engine, compression ratio 7.2/1, operated on leaded fuel, the compression ratio increased 0.35 units after light-duty deposit buildup and increased 0.19 units after heavy-duty deposit buildup. Deposit specific gravity was 4.0 and 5.5 after light- and heavy-duty deposit buildup respectively.

The octane-requirement increase due to thermal effects seems to us exceedingly high in the Waukesha single-cylinder engine as compared with multicylinder engine data, even when the magnifying effect of retarded spark is considered. Fig. B shows octane requirements at mbt spark obtained in a multicylinder L-head engine with clean combustion chambers at 150 F and 200 F cooling-water-out temperature superimposed on Mr. Warren's Figs. 7 or 9. It could be argued that a smaller or major part of the difference is due to the much higher octane level. We know that the octane scale is inadequate to compare accurately differences in knock tendency in terms of octane numbers at different octane levels.

It seemed to us interesting to apply Mr. Warren's analytical reasoning to the effects of spark retarding on octane requirement and power output after deposit buildup. In Fig. C are plotted octane requirement, power output, and airflow versus spark advance of a V-8 1953 competitive engine, compression ratio approximately 8.0/1. Deposits were built up on the dynamometer on leaded fuel in a heavy-duty schedule.

As shown in Fig. C the clean-engine airflow and the airflow decrease due to deposits are practically constant as

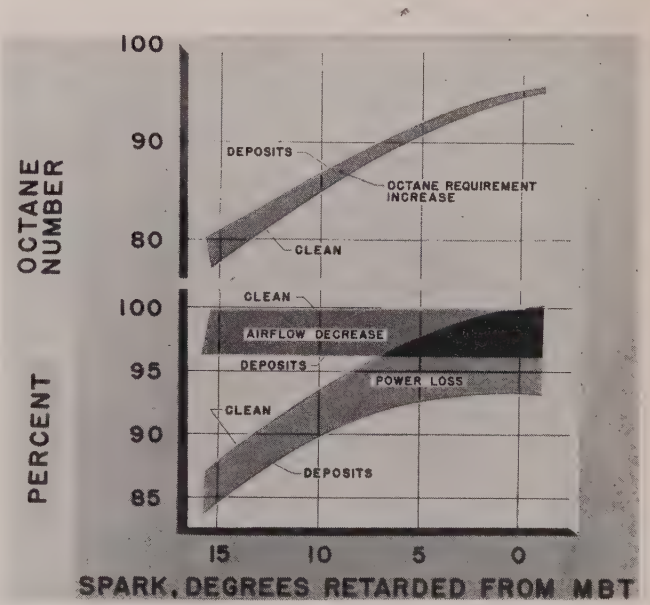

Fig. C – Deposit effects on octane requirement, airflow, and power at retarded spark, heavy-duty deposits, 1000 rpm, overhead-valve engine, approximately 8/1 compression ratio

the spark is retarded. This is not an individual result. We found the same or a very slight increase in airflow as the spark is retarded in many tests. In analyzing these results it can be reasoned that at retarded spark two effects oppose each other: a decrease in air heating due to lower combustion temperature and an increase in air heating due to the thermal effects of afterburning. Both effects might cancel each other out.

As shown further in Fig. C the octane-requirement increase due to deposits becomes greater as the spark is retarded. This is the usual experience and shared by Mr. Warren, but the reason seems not very obvious. Air heating is not greater at retarded spark, as has just been shown, and the heat-capacity effect of the deposits during compression is expected even to be lower. The only possible explanation we can see for this phenomenon is the inadequacy of the octane scale.

Contrary to octane-requirement behavior, the power loss decreases as the spark is retarded. Also this we find confirmed in many other tests. Power output is not governed by end-gas temperature but by the Btu's of the intake charge and the heat and mechanical losses encountered in their conversion to power. Thus, the following effects of deposits on power can be visualized:

1. Their thermal effect on the intake charge, which is power decreasing.
2. Their volume effect, which is power increasing but of secondary order.
3. Their thermal-insulating effect, which is power increasing.
4. Their heat absorption, which is power decreasing.
5. Their surface effect on mixture and gas flow, which is difficult to guess.

Under normal operating conditions airflow is considered the dominant factor. Roughly 70% of the power loss usually can be accounted to the reduction in airflow. Some of the other factors just mentioned are believed to make up for the balance.

At retarded spark, however, the power loss due to deposits decreases while the airflow of the clean engine and also the decrease in airflow due to deposits remain constant. Thus, in retarding the spark we reach a point where the per cent power loss equals the per cent air flow decrease and becomes even slightly lower. Trying to analyze these results we have obviously to eliminate the airflow as the cause of decreased power loss at retarded spark. We could reason that the thermal insulating effect is smaller during combustion due to lower temperatures and higher during expansion due to afterburning. If we discard surface effects, which if any might not change greatly, it seems likely that a significant decrease in heat absorption by the deposits is the dominant power-loss-reducing effect at retarded spark.

It seems to us that the author's findings provide a very good background for further investigations along these lines.

Whither the European Automotive Diesel?

Discussion continued from page 581

between 290 and 300 ton-miles per gal have been measured with a Krauss-Maffei engine, a figure never attained with any 4-stroke-cycle road vehicle in our knowledge.

On the basis of the situation as it exists today, Mr. Pitchford can make a good case for the 4-stroke-cycle automotive diesel. As to the future, he mentions a number of European makers of 4-stroke engines with no prospect of the 2-stroke amongst a single one of them. The authors wish to state that, at least with Alfa-Romeo, this is not the case – and this statement is based on more than a guess.

Mr. Pitchford's dim view on the road fuel consumption is contradicted by the preceding discussions.

The " acceleration" smoke of the 2-stroke-cycle engine is comparable to that of the turbocharged 4-stroke-cycle engine. Generally, it is tolerated, being of a short duration. It is due to the engine getting rid of soot deposits accumulated in the exhaust system during the preceding idling. Steps to correct this annoyance have already achieved measurable success.

The service difficulties with 4- or 6-hole nozzles should, with the present fuel filters, not be overestimated. Most direct-injection 4-stroke engines are in the same boat. For his smallest size engines List uses swirl chambers with equal success as the 4-stroke engines employing this combustion system. The choice depends in both cases largely on the cylinder size.

The authors humbly acknowledge that the title of the paper is somewhat misleading and offer their apologies for it.

Professor Lichty favors the short-stroke engine. The authors readily agree to the advantages enumerated. There are, however, points against the short-stroke engine also. In case of a 2-stroke-cycle engine, one of them is that the short-stroke engine requires for scavenging a higher blower pressure, as the following reasoning will show.

For a given delivery ratio and piston speed:

$$\frac{V_{del}}{(D^2\pi/4)s} = C_1 \text{ and } ns = C_2$$

The amount of air going through the ports per cycle is, with good approximation:

$$V_{del} = \frac{C_3 A \sqrt{p}}{n}$$

where p is the scavenge pressure and A the mean area of the ports, the height of which is proportional to the stroke and the width to the bore, $A = C_4 s D$.

From these, by elimination:

$$C_1 \frac{D^2\pi}{4} s = C_3 C_4 s D \sqrt{p/n} = C_3 C_4 s D \sqrt{p} \frac{s}{C_2}$$

or

$$\sqrt{p} = \frac{C_1 C_2}{C_3 C_4} \frac{\pi}{4} \frac{D}{s}$$

or

$$p = C_s \left(\frac{D}{s}\right)^2$$

The large borestroke ratio engine requires a higher scavenge pressure under otherwise similar circumstances. This results in less net power output and higher fuel consumption. Thus, 2-stroke-cycle very-short-stroke engines are not economical.

The New Ford V-8 Engine

Robert Stevenson, Ford Motor Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1954.

IN 1948, our company decided to convert its engine plant facilities to accommodate modern transfer equipment and more complete automation. Our L-head V-8 engine was performing satisfactorily on fuels available at that time, and there was every reason to believe that it would continue to do so well beyond the time when the new tooling became available. However, since costly major tooling expenditures require a minimum of eight years for amortization, it was decided that a new engine would best meet the requirements of future improvements in fuel and higher compression ratios.

Consequently, the engineering staff was asked to design a new engine that would meet these requirements as well as those of the vehicle planned for release in 1954. For obvious reasons, it was also necessary to design into this new engine a high degree of adaptability, to permit future changes as technical knowledge increased or economic conditions dictated, without major changes in tooling or facilities. (See Fig. 1.)

In establishing the initial design concepts for this new engine we found that it was easy to decide on the number and arrangement of cylinders, since the V-8 had already earned world-wide acceptability in our vehicles since 1932. Coupled with these many years of V-8 manufacturing and engineering experience was the comfortable knowledge that an ever-increasing number of manufacturers have recognized the inherent advantages of the V-8 engine, with its short, rigid structure and characteristic smoothness at all loads and speeds.

The other major design aims were:

1. Displacement, package size, and weight to be approximately the same as in the L-head V-8.
2. Greatest possible structural rigidity, to maintain a high standard of smoothness with probable future compression ratio increases.
3. A very short stroke, to increase mechanical and thermal efficiencies.
4. An overhead-valve arrangement, to permit maximum volumetric efficiency, if required by future increases in compression ratio.
5. Maximum mechanical simplicity to allow for traditional Ford manufacturing economies.

In addition to these major design considerations, there were many other objectives that had to be met, such as: lower valve temperatures, rotating valves, full-flow oil filtration, thorough engine ventilation, a single water pump, chain-driven camshaft, combustion chambers with maximum quench areas, and, of course, more horsepower and torque.

During the entire period of development, which consumed nearly 600,000 design man-hours, we maintained a close liaison with our manufacturing staff to make sure that our new engine would be

ALTHOUGH the new Ford V-8 engine has the same displacement and compression ratio as the old V-8 does, the new engine develops 130 hp and 214 ft-lb maximum torque to only 110 hp and 196 ft-lb torque for the old. Specific fuel consumption is also lower in the new engine over the entire speed range.

The new engine has a very short stroke, to increase mechanical and thermal efficiencies. It also has overhead valves to permit maximum volumetric efficiency to be attained, if required by future increases in compression ratio.

The Author

ROBERT STEVENSON (M '44) is chief engine engineer of Ford Motor Co. He was educated at the University of Michigan and joined Ford in 1934. Mr. Stevenson has held his present position since 1952.

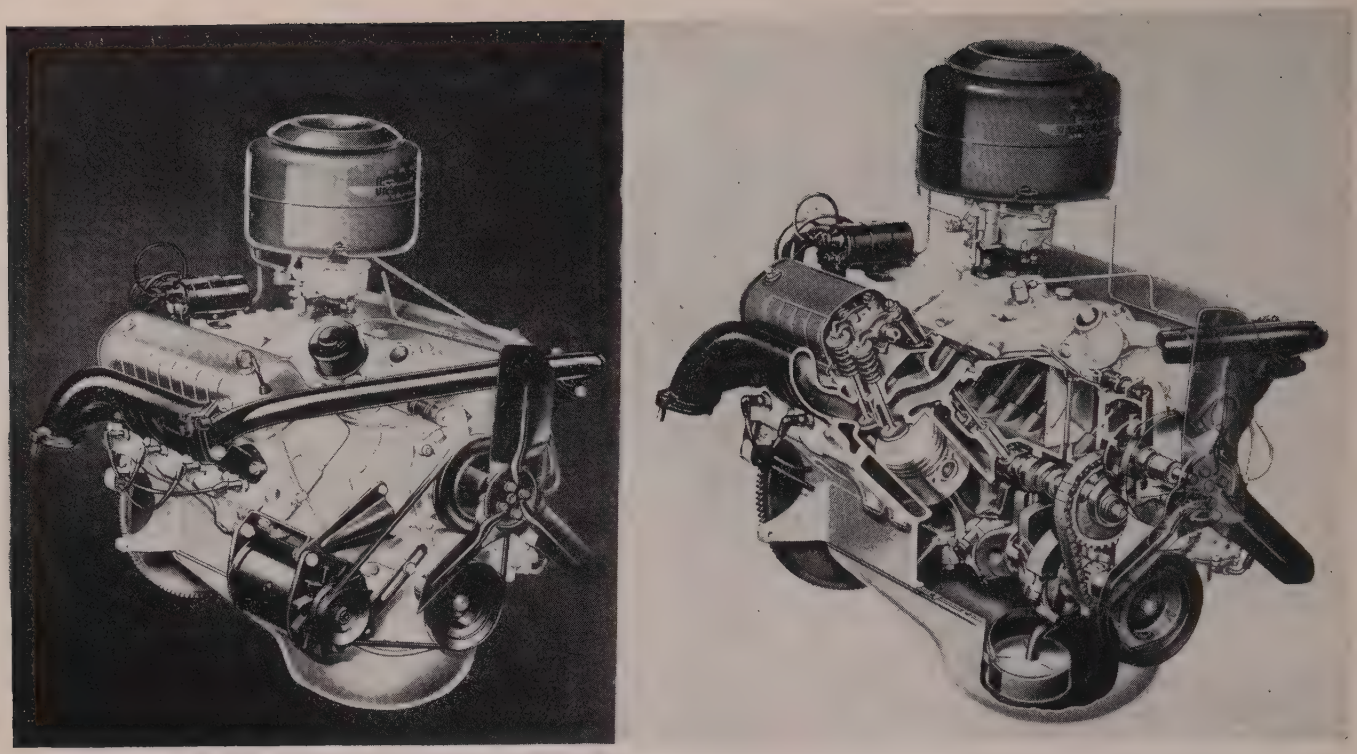

Fig. 1 – 130-hp Ford V-8 engine

compatible with the latest types of mass-production transfer equipment. It is a tribute to the manufacturing staff that we were able to accomplish that objective without a single sacrifice of our design principles.

Cylinder Block Structure

The cylinder block in the new V-8 engine is a one-piece casting of high-grade iron alloy, designed for unusually high strength and great rigidity. Crankcase skirts are carried well below the crankshaft centerline to provide the stiffness essential to smooth performance in a high-compression, high-output engine. Fig. 2 shows the new block with the front corner cut away to reveal one of the five bulkheads, which stiffen the structure and serve as supports for the crankshaft and camshaft bearings. Cylinder-head attaching bolt bosses are formed in the outside walls to prevent cylinder-bore distortion when the heads are pulled down tightly. Cylinder bores are attached only at the top and bottom decks, and are completely surrounded by the cooling water over their entire lengths.

Casting of the new cylinder block is made much easier because of the overhead-valve construction, as can be seen by comparing Figs. 3 and 4. The number of cores required has been reduced from 29 to 14, making the new block more compact, as well as simplifying its construction.

Fig. 5 compares front views of the 1953 L-head V-8 cylinder block, and the new block, and illustrates the effectiveness of the deep-skirted crankcase in providing additional support for the 5-bearing crankshaft in the new engine.

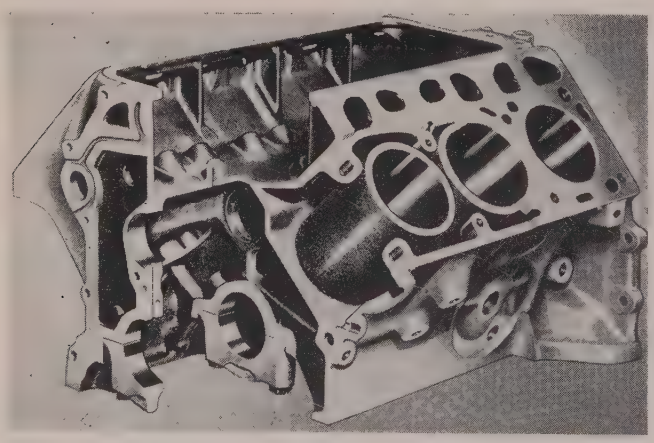

Fig. 2 – Cylinder block was designed for maximum structural rigidity

Fig. 3 – 29 cylinder block cores required for 1953 V-8

Fig. 4 – 14 cylinder block cores required for 1954 V-8

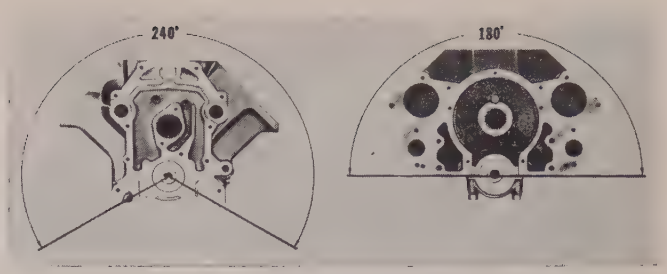

Fig. 5 – 240-deg main bearing support

A shallow, more rigid oil pan, with a continuous sealing flange, is an extra dividend provided by the deep-skirt crankcase, as illustrated in Fig. 6. The sealing flange is 1 in. wide and has parallel beads stamped into its surface. This construction makes it unnecessary to provide additional reinforcement, and the beads are so located that they form stiff sealing bridges between the bolt holes. The sump is located at the front of the engine because of the location of a cross-member in the 1954 front suspension. The rounded nose of the sump has the ability to deflect stones, with less chance of sustaining damage than one having a rectangular section. An antisurge baffle is incorporated in the oil pan, which is designed to supply oil to the externally mounted pump on grades as steep as 30%. The drain plug is located at the rear of the sump.

Our research during the development of this engine revealed that one of the critical structural areas in high-output engines is the method of attachment to the flywheel housing and drive-line components. We found that an engine and transmission assembly could sag as much as 1/16 in. with the conventional attachment. This led to the development of the wide-base bell housing connection (shown in Fig. 7) in which both sides of the cylinder block flare out widely at the rear to provide attaching points outside the flywheel perimeter and nearly 3 in. below the centerline of the crankshaft. With this kind of attachment, the tendency to sag is minimized, assuring good alignment of the crankshaft with the drive line.

A 3-point starter mount (Fig. 8) is used in this new engine. The attaching bolt bosses are cast into the flywheel housing and the starter is bolted directly to the housing. A close-fitting recess is provided in the cylinder block, eliminating the necessity for a cover plate at this point. A neoprene seal closes the space between the starter and the block and prevents metal-to-metal contact between the two.

Design Details

The crankshaft in the new V-8 has five main bearings instead of three, as formerly, and is precision-molded of alloy iron. Cast crankshafts

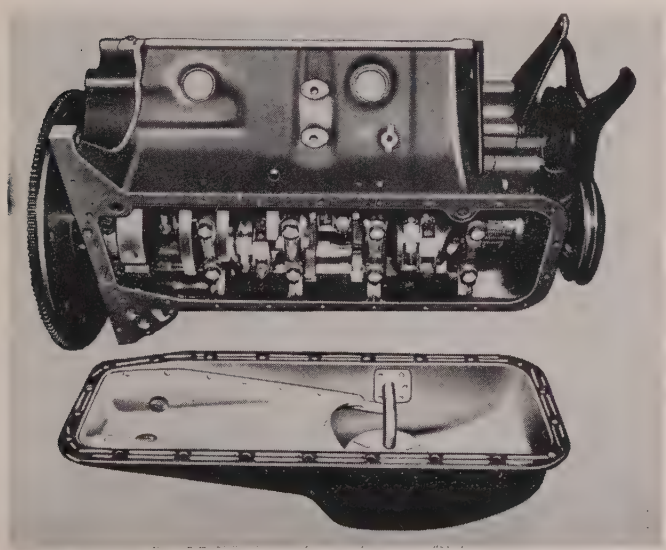

Fig. 6 – Continuous oil pan seal

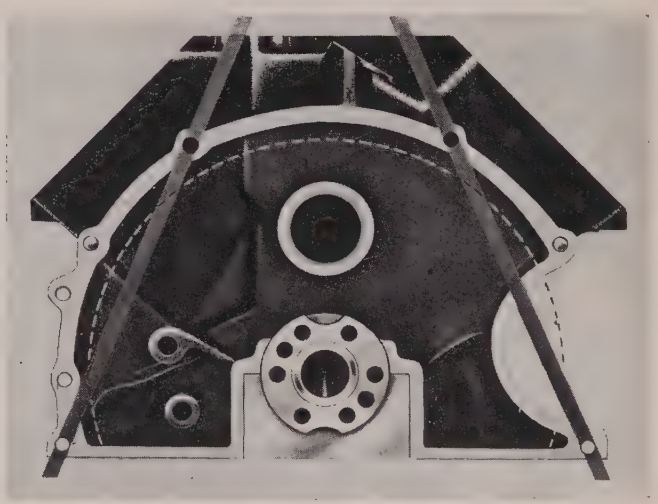

Fig. 7 – Wide-base bell housing connection

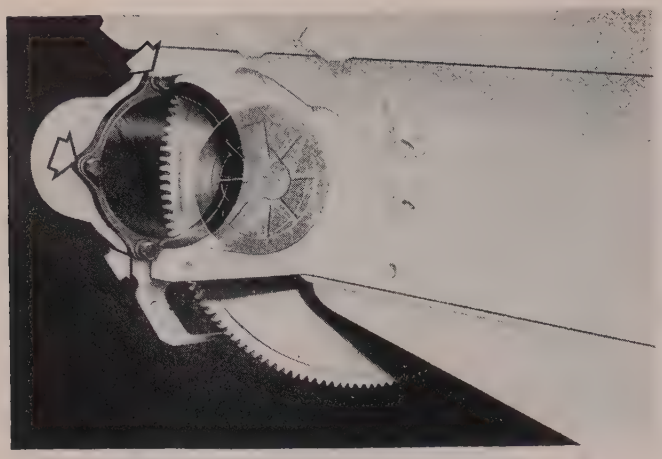

Fig. 8 – 3-point starter mount

Fig. 9 – Comparison of crankshaft for 1953 engine with crankshaft for 1954 engine

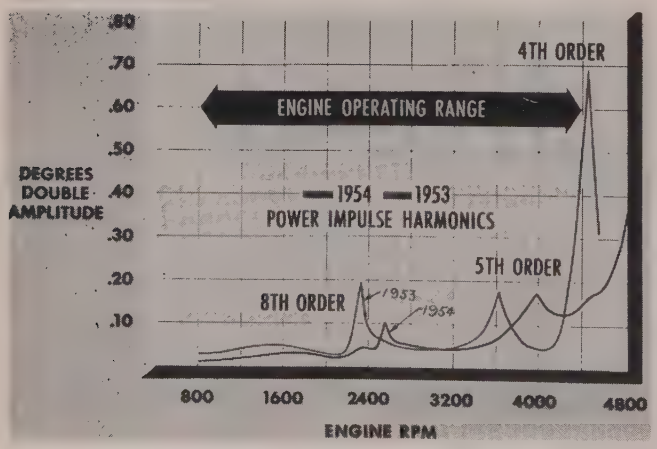

Fig. 10 – Reduced torsional vibration

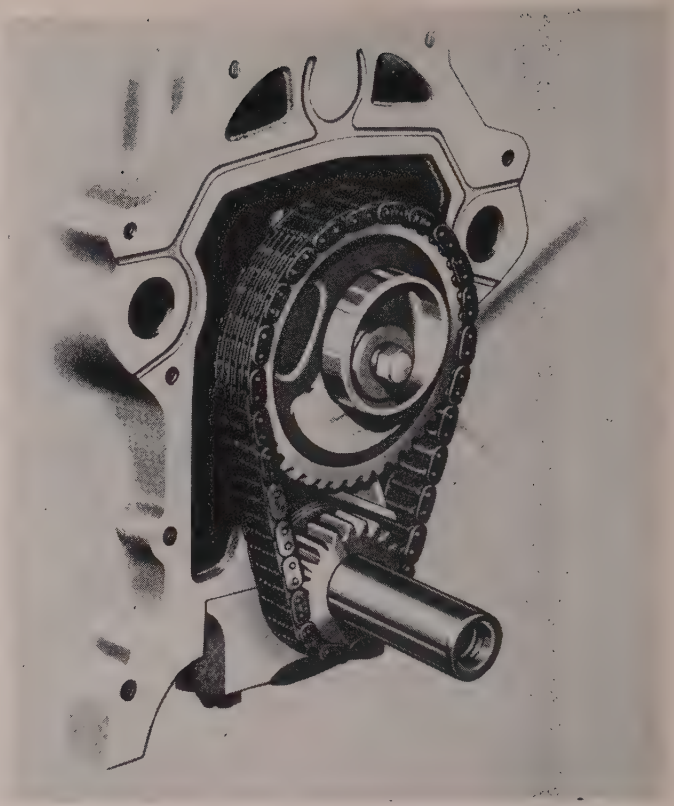

Fig. 11 – Chain-driven camshaft

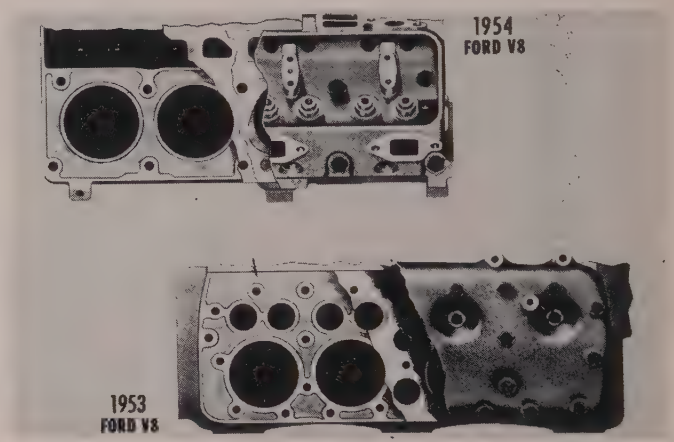

Fig. 12 – Comparison of 1953 and 1954 cylinder heads

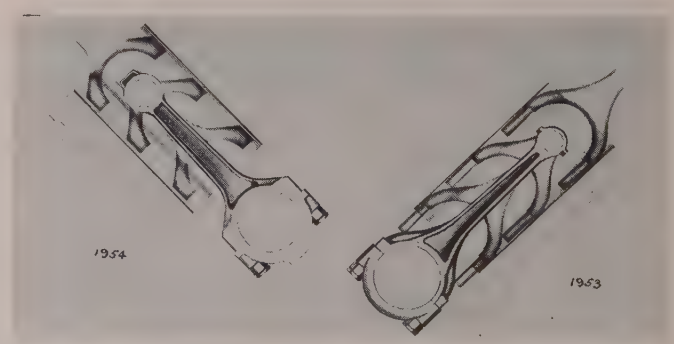

Fig. 13 – New Engine (left) has larger bore, which allows separate bolts to be used in connecting rods, rather than integrally forged bolts used previously

are exclusive with our company, and this new shaft can have eight integral counterweights because casting is not limited to a single plane, as is the forging process. The very short stroke in this new engine (3.10 in.) results in a crankshaft having a large amount of journal overlap – nearly 13/16 in., as shown in the lower part of Fig. 9. This contributes greatly to the lightness and rigidity of the new shaft, which is actually 16 lb lighter than its L-head counterpart.

The connecting-rod journals are 2 3/16 in. in diameter and the bearings have a total effective area of 39.08 sq in. The main bearing journals are 2½ in. in diameter and the total effective area is 28.28 sq in. End thrust is taken on a flanged center main bearing. Steel-backed microbabbitt is used for both main and connecting-rod bearings.

The torsional vibration studies shown in Fig. 10 also reflect the stiffness of the new shaft. Harmonic peaks are not only lower in amplitude, but they occur at higher engine speeds, fourth-order vibration being well outside the engine operating range. For these reasons, it was unnecessary to include a torsional vibration damper on the new engine, in spite of its greater output.

The precision-molded, special alloy camshaft is designed with lobe and ramp contours, which result in quiet operation and long life. As in previous Ford engines, the timing has been developed for good low-speed torque without sacrificing performance at highway speeds. The cam lobes are phosphate-coated and are not hardened after casting. To provide for tappet rotation, the lobes are slightly tapered and the tappets are ground with a large spherical radius on the foot. A great deal of effort went into developing metallurgical compatibility between the cast shaft and the cold-headed tappets.

As illustrated in Fig. 11, the camshaft is driven by a 1-in. silent chain, having a three-eighths pitch. Deflections are held to a minimum by the short center distances between sprockets. The camshaft sprocket is hardened cast iron, and the crankshaft sprocket is alloy steel, case-hardened.

The fuel pump is driven by the hard chrome-plated eccentric, which is bolted to the front of the camshaft sprocket. As an indication of the extra attention that was paid to small details, you will note that the eccentric is provided with a counterbalance.

Fig. 12 shows a comparison of the 1954 and 1953 cylinder heads. The new cylinder head is much more rigid than that used in the previous engine, because of its much deeper cross-section. This permits a reduction in the number of holddown bolts from 24 to 10. The bolts are located closely around the cylinder bores in a balanced arrangement, which allows the use of a thin, all-steel cylinder-head gasket for maximum heat transfer between the cylinder head and block. This illustration also shows clearly how the location of bolt bosses along the walls of the cylinder block frees the cylinder bores from the distorting effects of connecting webs.

As in the past, connecting rods and caps are separately forged. This technique permits us to design the critical areas in the cap for high strength without an excessive weight penalty. Balancing lugs are located at each end of the rod assembly.

Separate bolts are used in the new connecting rods, providing a stronger connection than is possible with the integrally forged bolts previously used. Fig. 13 shows that it is the larger bore in the new engine that permits this improvement. The new connecting-rod assembly is shown in Fig. 14. It will be noted that a unique cam-headed bolt is fitted into an eccentric counterbore in the rod to prevent the bolt from turning. This avoids weakening the rod structure at the most critical point by notching it for the commonly used T-head bolt.

In the new engine, the use of autothermic aluminum-alloy pistons is continued. However, as shown in Fig. 15, only three rings are employed – all above the piston pin. The oil control ring is equipped with an expander for better oil control throughout the life of the engine.

Not so many years ago, piston and gasket destruction tests of 50-hr duration were considered satisfactory. Today, however, our standard has

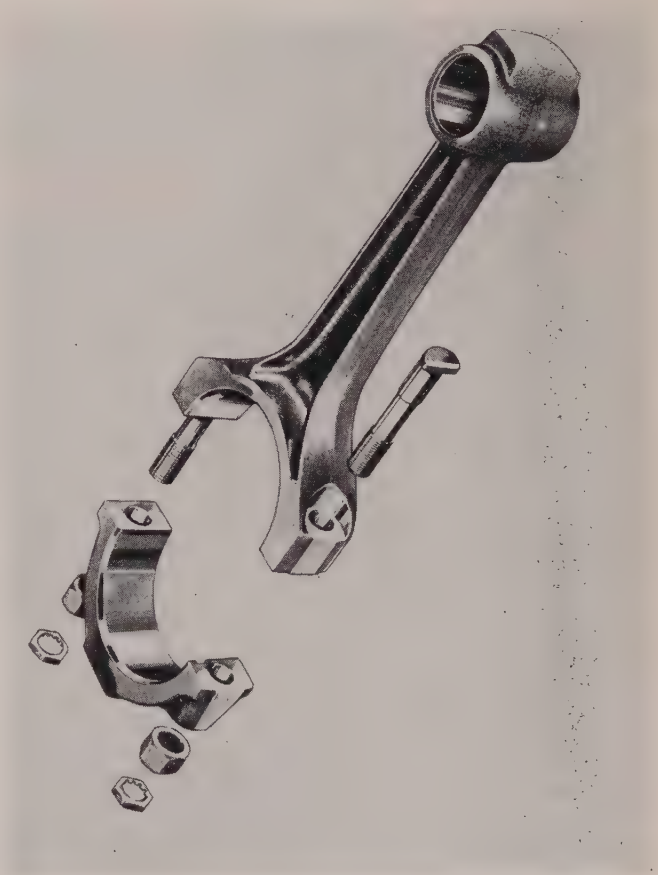

Fig. 14 – Separate connecting-rod bolts

Fig. 15 – Comparison of 1953 and 1954 pistons

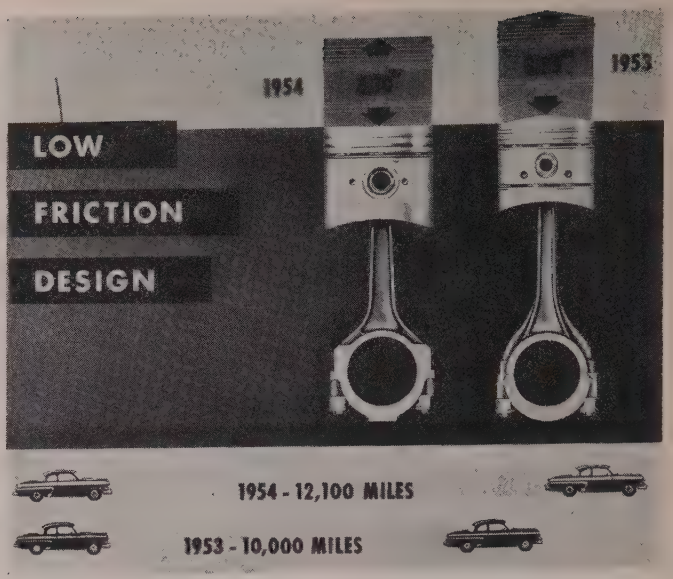

Fig. 16 – Short stroke of new engine means 17% less piston travel

been raised to a minimum of 200 hr. In this test, the engine is run at wide-open throttle at peak torque speed, with the spark advanced 15 deg beyond the normal mbt setting, resulting in heavy, continuous detonation.

The new engine has a 3.5-in. bore and a 3.1-in. stroke, as compared with a 3.19-in. bore and a 3.75-in. stroke, in the L-head V-8. The difference is shown in Fig. 16. It results in a reduction of 17% in piston travel. In Fig. 17, this reduction is translated into extra car-miles with a given amount of piston travel. Since the new engine has approximately the same displacement and compression ratio, we believe that this low-friction design is the greatest factor contributing to the higher power output of the 1954 Ford V-8.

This is further substantiated by the comparison of bmep and fmep curves shown in Fig. 17. You will note that in the L-head V-8, fmep was 33 psi at 4000 rpm, whereas in the new engine it is only 26 psi. The bmep is correspondingly greater over the entire speed range, and reaches a maximum of 135 psi at 2000 rpm.

Another advantage of the larger bore and short stroke is shown graphically in Fig. 18, which pictures a hypothetical cylinder of the same displacement as that of the new engine, but with a much smaller bore. The smaller bore is seriously limited in valve diameters that could be used. All other things being equal, the larger bore permits the use

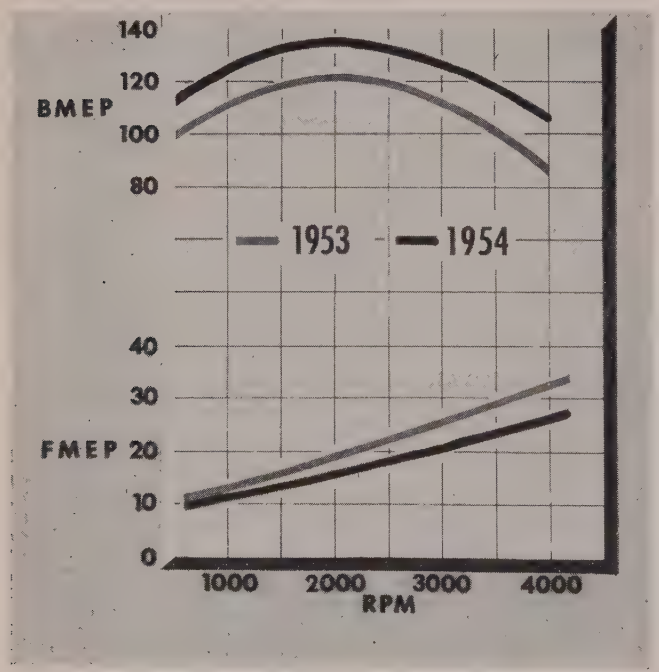

Fig. 17 – Bmep and fmep for 1953 and 1954 engines

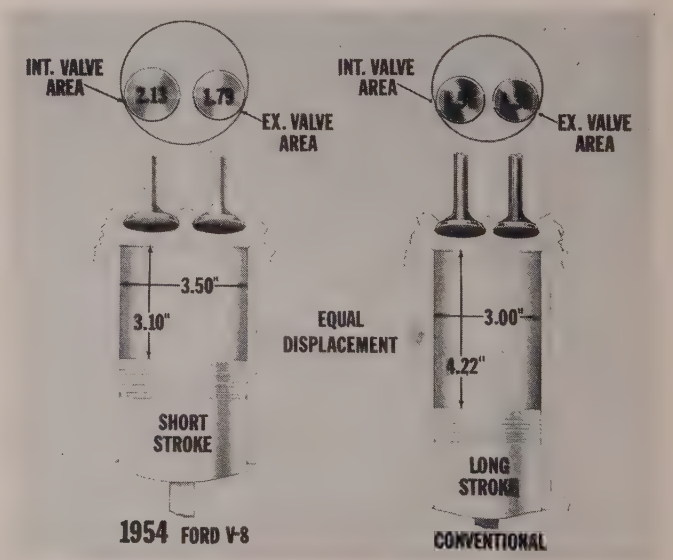

Fig. 18 – Larger bore equals room for larger valves

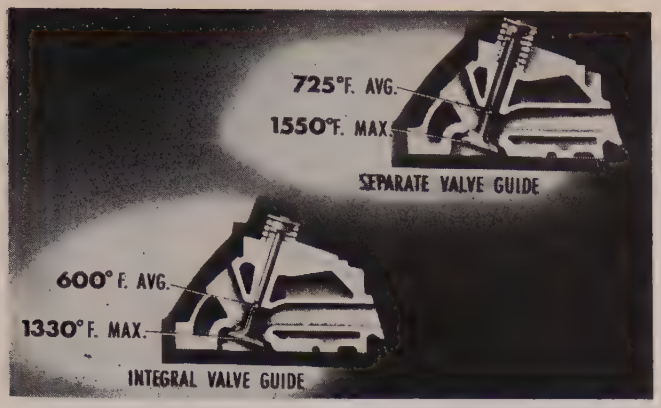

Fig. 19 – Integral valve guides

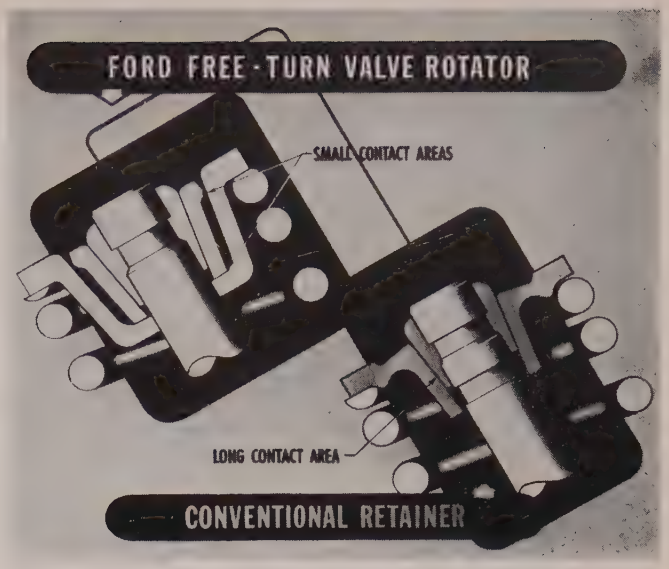

Fig. 20 – Free-turn valve rotator compared with conventional retainer

of valves with head areas 36% larger, providing obvious breathing advantages.

One of the major design objectives was to reduce valve operating temperatures, which have become an increasingly critical problem in the industry as compression ratios have been increased. Since the greatest cooling effect is realized through the valve stem into the coolant, our attention was focused on this area. As a result, we have achieved a substantial reduction in critical valve temperatures through the use of integral valve guides, as illustrated in Fig. 19. Tests were conducted with and without separate guides in the same cylinder heads, and the results are shown in the illustration. Average stem temperatures were reduced from 725 to 600 F, and maximum valve-head temperatures from 1550 to 1330 F. These reductions were possible because, with integral guides, there is only a clearance and one wall between the valve stem and the coolant, whereas with separate guides, heat must penetrate twice as many insulating barriers.

All valves in the new engine are equipped with our own design of free-turn valve rotators. In Fig. 20, these are pictured at the left. The conventional valve spring retainer (on the right) comprises a spring retainer washer and two tapered keys. In the new design, however, the two retainer keys are contained in a tapered sleeve. The spring retainer washer bears against this sleeve. This is done to reduce the friction moment, which resists turning of the valve. This reduction in friction moment means that the valve, under the influence of slight vibrations that exist in all valve springs, will begin to turn at a much lower engine speed. This has resulted in a material increase in valve life for both intake and exhaust valves.

The cooling system in the new engine is similar to that of the previous engine in that the series-flow principle is employed. However, the new system has a single pump instead of two, as shown in Fig. 21. This single, high-capacity pump discharges directly into an equalizing chamber from which a balanced flow is fed to both cylinder banks.

The L-head V-8 engine required a very large

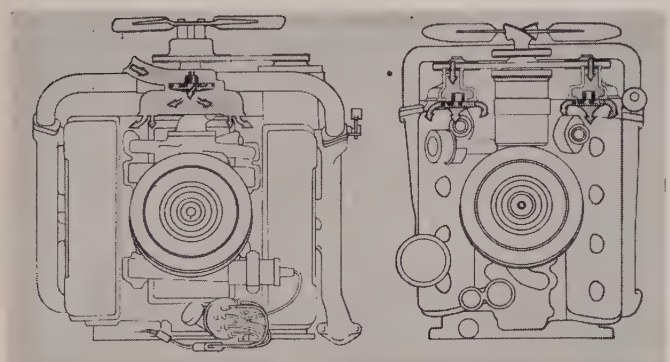

Fig. 21 – 1954 engine (left) has one water pump; 1953 engine (right) has two water pumps

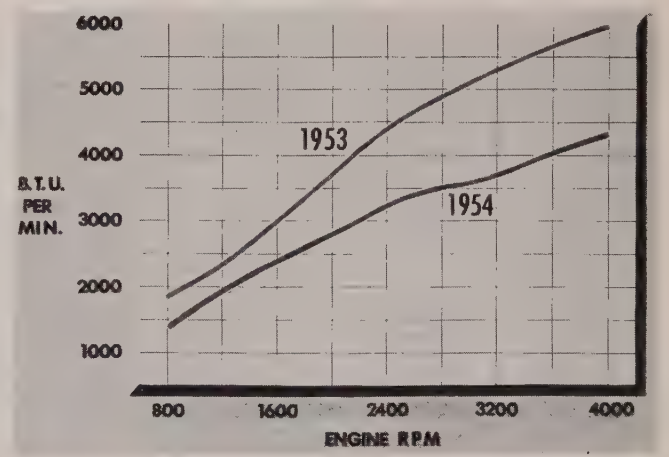

Fig. 22 – Full-load heat rejection to coolant

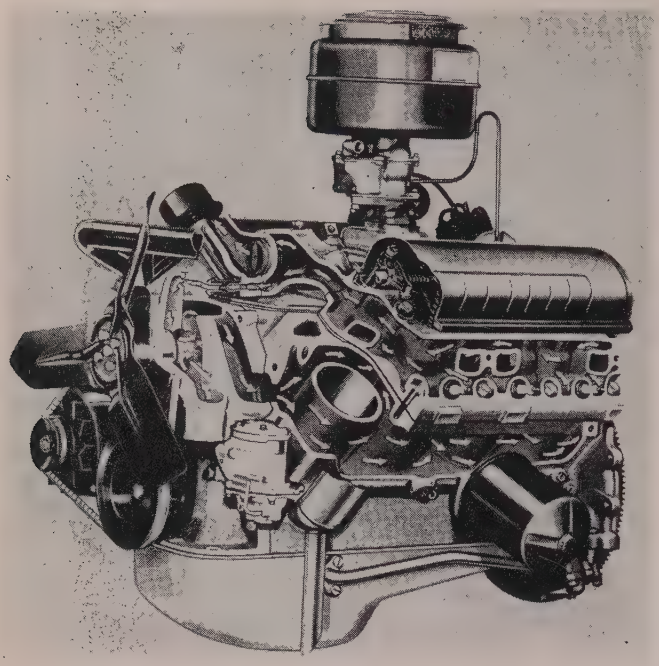

Fig. 23 – Cooling system

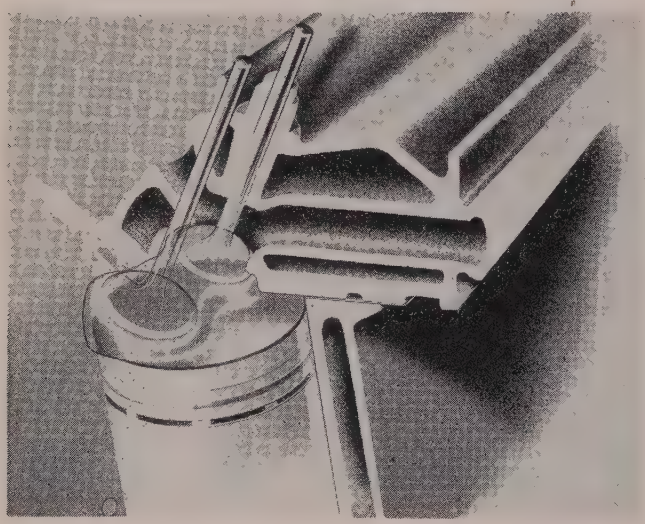

Fig. 24 – High compression

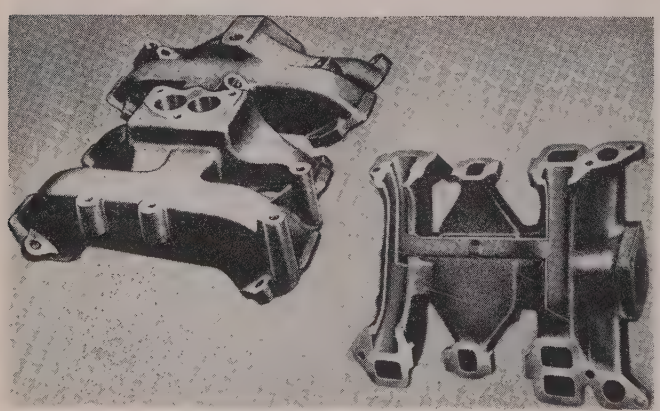

Fig. 25 – Balanced breathing

cooling capacity because of the long exhaust passages cast into the cylinder block, resulting in high heat rejection to the coolant. Fig. 22 shows that the heat rejection of the L-head V-8 amounted to 6000 Btu per min at 4000 rpm (wide-open throttle), whereas the new engine shows a reduction to 4300 Btu per min at the same speed.

The new cooling system is illustrated in Fig. 23. From the equalizing chamber, the coolant is directed through the cylinder block to openings in the rear portion of the cylinder heads, where it reverses its direction and flows through the large passages surrounding the combustion chambers, valve guides, and spark plugs. The very short exhaust passages are effective in keeping the heat rejection to a minimum. The coolant is then returned to the radiator through a passage in the front of the intake manifold leading to a common outlet connection. A single cartridge-type thermostat is contained in the outlet connection, and a fixed bypass is provided for quick warmup. This system assures uniform cooling to each bank of cylinders, with a coolant balance within 2 deg at all speeds and temperatures.

Extensive testing of all types of combustion chambers during this engine program has substantiated the fact that a highly turbulent, large-quench-area design was the most efficient. It permitted a more homogeneous mixture of fuel and air, an optimum spark-plug location, side-by-side valves, which meant a short, compact valve train with low inertia and less deflection. It scavenged exhaust gases thoroughly and kept combustion-chamber deposits to a minimum.

As employed on our engine there is no evidence of serious harshness or roughness, no doubt due to the emphasis we have placed on extreme structural rigidity throughout the complete powerplant. This chamber is illustrated in Fig. 24.

The intake manifold (shown in Fig. 25) incorporates over-and-under passages, through which we were able to obtain the largest possible port areas within the space dictated by the engine length. Risers for the duplex carburetor are connected independently to longitudinal passages feeding lower ports at one end and upper ports at the other end. The branches meet these longitudinal passages at right angles, so that the fuel-air mixture is divided equally to both sides. Contours of the passages are smooth and equal in cross-section area throughout, and generous radii are provided at all junctions. Exhaust heat is diverted through the central passage surrounding the carburetor inlets during engine warmup. This is accomplished by a flapper-type thermostatic valve, located at the entrance to the right-hand exhaust manifold.

Fig. 26 shows that, over the years, the L-head V-8 octane requirement has always been below the average octane number of available regular fuels. The new overhead-valve engine, while taking

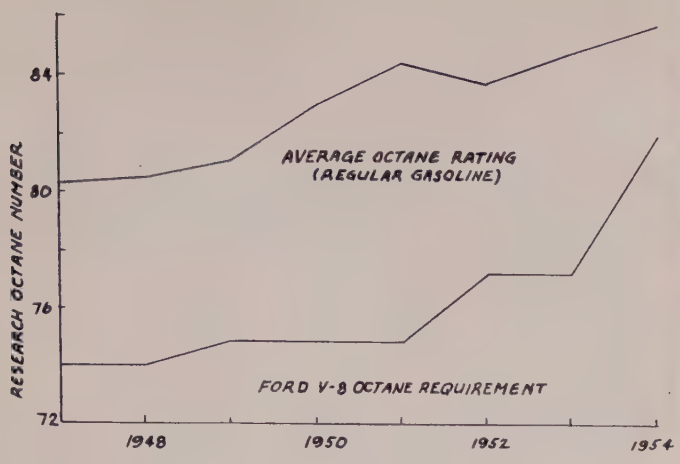

Fig. 26 – Octane trends

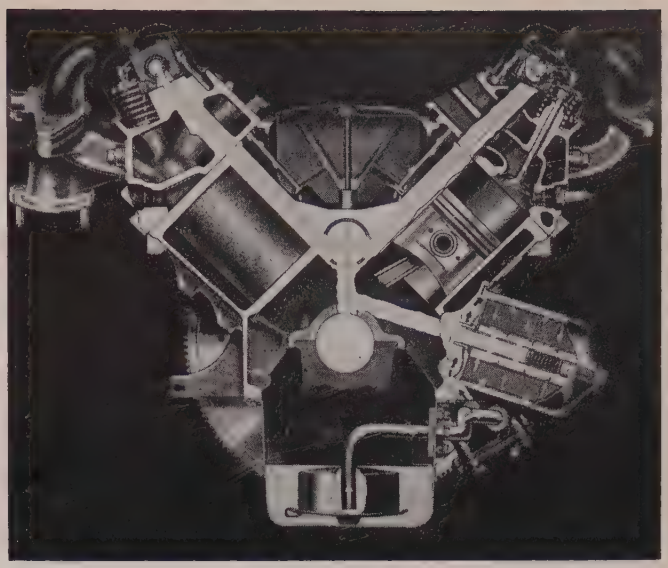

Fig. 27 – Lubrication system

greater advantage of the increasing octane numbers in today's fuels, continues the tradition of operating satisfactorily on regular gasoline.

Lubrication and Oil Control

The lubrication system of the new V-8 is illustrated in Fig. 27. As before, a full-pressure system is provided. Oil is picked up from the 4-qt sump through a fixed screened inlet and transferred to the externally mounted gear-type pump. The pump maintains a minimum pressure of 15 psi at idle, and an average of 40 psi at 40 mph. A bypass valve recirculates the oil whenever the pressure rises above 45 to 50 psi.

The oil is pumped through a short, drilled passage in the cylinder block directly to the full-flow oil filter. It leaves the filter through the hollow center bolt and is fed to the main oil gallery, which runs the full length of the cylinder block. Main and camshaft bearings are lubricated directly from this gallery. Connecting-rod bearings are lubricated by full pressure, through the crankshaft. The supply holes in the crankshaft index with holes in the connecting rods, providing lubrication for the cylinder walls by intermittent pressure stream, as shown in Fig. 28.

At the rear camshaft journal, indexing holes in the camshaft supply intermittent pressure lubrication to the distributor shaft pilot in the cylinder block (Fig. 29). The distributor gears are gravity-fed with oil returned from the rocker mechanism.

Oil pressure to the rocker shafts is reduced through the camshaft and camshaft bearings. The

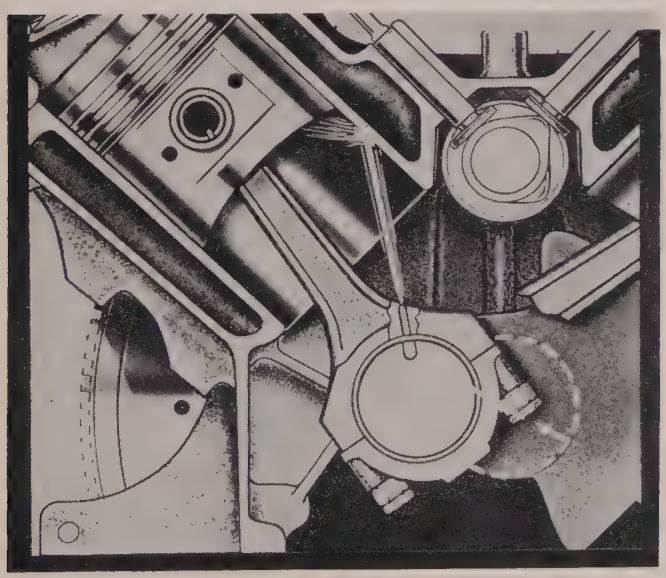

Fig. 28 – Cylinder-wall lubrication

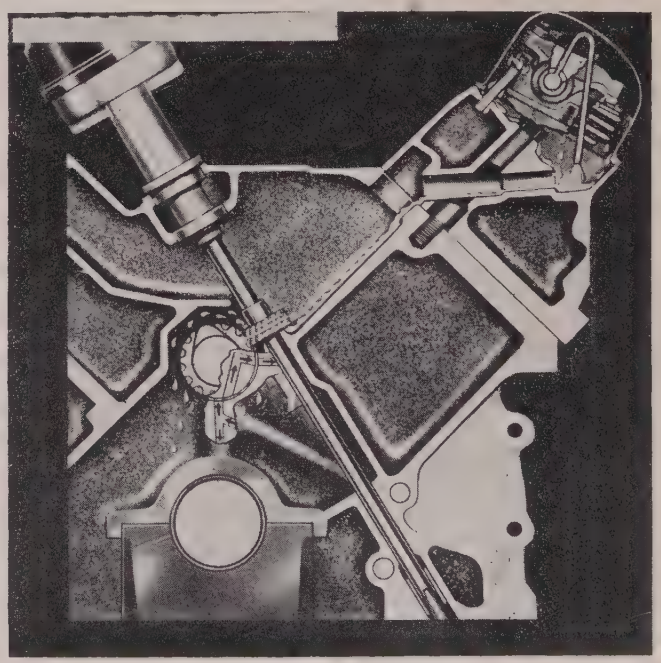

Fig. 29 – Distributor drive lubrication

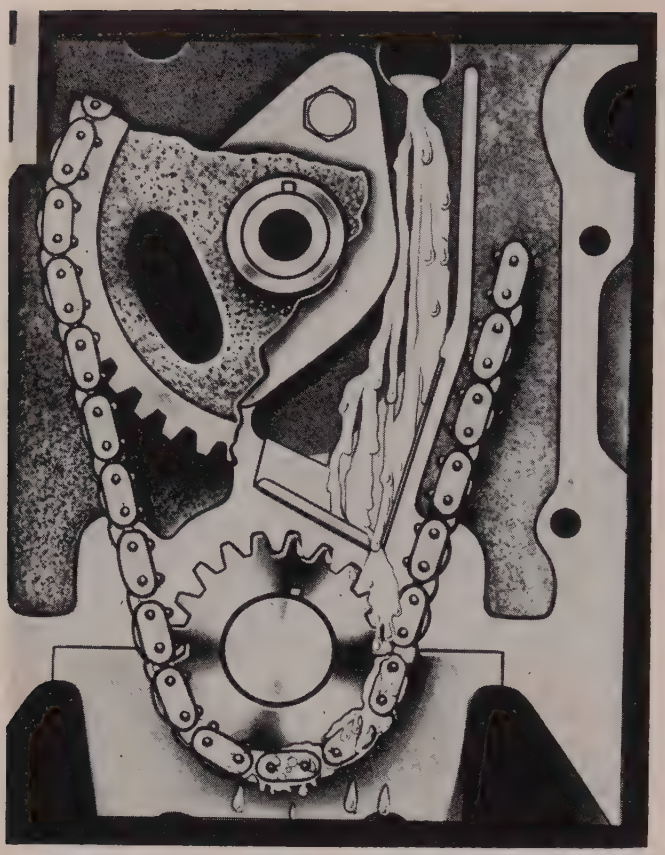

Fig. 30 – Timing chain lubrication

hollow rocker shafts act as oil galleries to feed the rocker arms. Holes in the rocker arms squirt oil onto the valve stems and push-rod sockets, from which it drains into the valve trough in the top of each cylinder head. Holes at the front and rear of the cylinder heads permit the oil to drain into the engine valley to lubricate the tappets, after which it returns to the sump through drain holes provided at the front and rear. At the front, the oil returning to the sump is directed onto the timing chain by a deflecting trough (Fig. 30).

Neoprene umbrellas (as in Fig. 31) are fitted to both intake- and exhaust-valve stems to control the amount of oil fed to the integral valve guides. The umbrellas move up and down with the valve, permitting oil mist to enter the guide when the valve is closed, but covering the entrance when it is open.

Fig. 32 illustrates the method used to assure good sealing of the rocker covers in the new engine. The inset shows how the gasket is compressed against the unmachined, slightly rounded ridge that extends around the top of the cylinder head. By this means, a high unit pressure is exerted against the gasket, providing a uniformly tight seal. The deep valve trough adjacent to the outside ridge keeps the gasket from being immersed in oil and prevents ultimate wicking.

The 1954 engine is equipped with a full-flow oil filter (Fig. 33), which cleans all of the oil before it can reach vital bearing surfaces. This cleansing is of greatest importance when the engine is new, since our tests have shown that a greater amount of engine wear occurs during that period. The full-flow filter protects the new engine by removing any grains of sand or metallic particles accidentally built into the engine.

Fig. 33 shows how engine wear was reduced in tests where both full-flow and partial-flow filters were run the equivalent of 20,000 miles. A mechani-

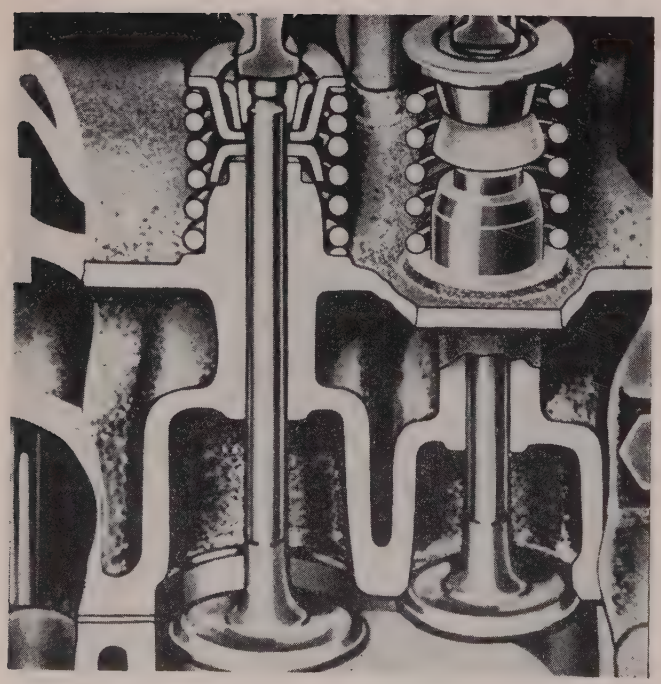

Fig. 31 – Valve-stem oil control

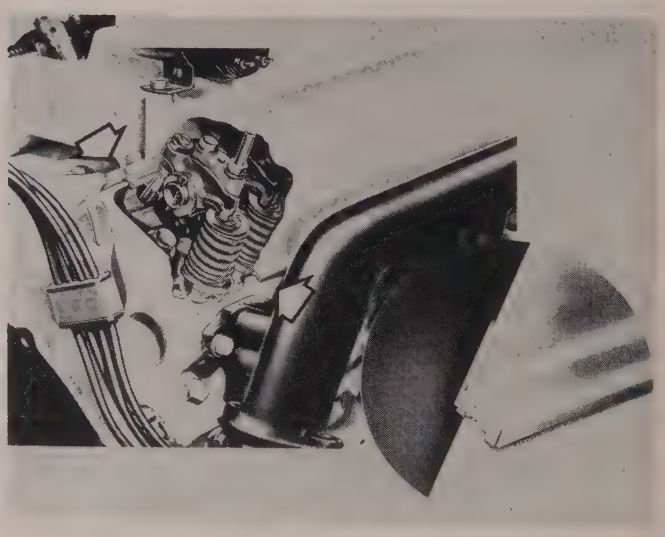

Fig. 32 – Rocker cover sealing

Table 1 – Brief Specifications for Ford V-8 Engine

Type	Overhead-Valve, 90-deg V-8
Bore and Stroke, in.	3.50 x 3.10
Displacement, cu in.	239
Compression Ratio	7.2 to 1
Maximum Gross Horsepower[a] at 4200 Rpm	130
Maximum Gross Torque[a] at 1800–2200 Rpm, lb-ft	214
Maximum Bmep at 2000 Rpm, psi	135
Compression at Cranking Speed of 120 Rpm, psi	125
Average Piston Speed at 4200 Rpm, fpm	2170
Taxable Horsepower	39.2

[a] Corrected to 60 F dry air at 29.92 in. Hg.

cal means of wear reduction, such as the full-flow filter, lessens the need for reliance on qualities of lubricants, which are outside the control of the engine designer.

A metal diaphragm at the base of the filter prevents oil from draining back into the sump when the engine is stopped, assuring an immediate oil supply to the bearings when it is started again.

Extensive research into engine ventilation, which involved development of special instrumentation to measure airflow, resulted in the constant-flow ventilation system used in the new V-8 engine. By careful design, it was possible to achieve an air flow of from 1 to 4 cfm over the entire speed range.

As shown in Fig. 34, fan-driven air enters at the top of the engine through an oil-wetted filter. After ventilating the rocker covers, the air is carried downward into the crankcase, where it picks up corrosive blowby gases and is immediately discharged through a screened road draft tube. The outlet is baffled and has a drain-back tube to prevent oil pull-over.

Fig. 35 shows a direct comparison between the old and the new V-8 engines, which is all the more significant because the two engines have the same compression ratio and displacement. Whereas the L-head V-8 developed a maximum of 110 hp and 196 lb-ft torque, the new engine develops 130 hp and a maximum torque of 214 lb-ft. Specific fuel is also lower in the new engine over the entire speed range. Table 1 gives the specifications for the engine.

In developing this new engine, more than 400 experimental engines had to be built, using the ordinary tool room equipment available in our experimental shops. These engines were tested on dynamometers for a total of more than 160,000 hr, and were installed in vehicles that accumulated more than 2½ million miles. These experimental engines were rebuilt as many as eight times in order to test individual components as they were developed. As a result of this extensive and thorough testing, we are introducing the new Ford V-8 engine with every confidence that it not only meets our design objectives, but that it will establish for itself a reputation as enviable as its famous predecessor.

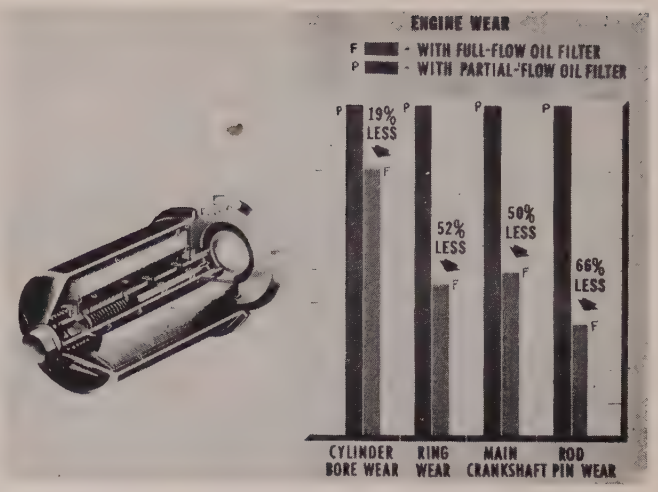

Fig. 33 – Full-flow oil filter – 20,000-mile oil filter tests

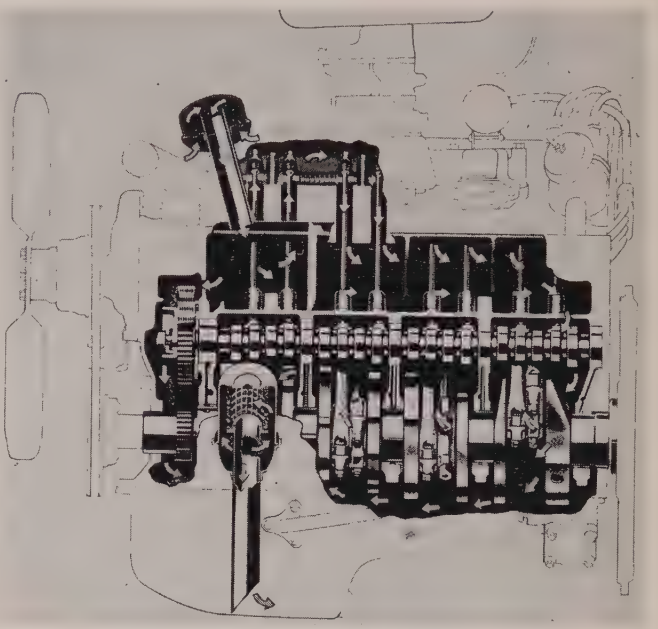

Fig. 34 – Engine ventilation

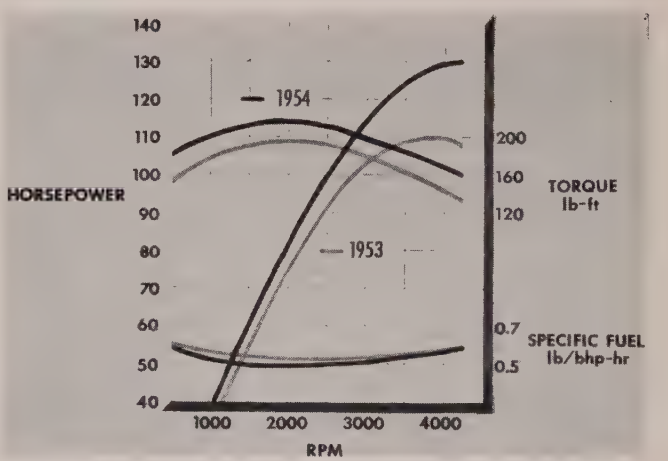

Fig. 35 – Performance curves for 1953 and 1954 engines

The Evaluation of the Energy Released

THE combustion process which occurs in the internal-combustion engine has been studied for many years employing various means of investigations. The more important means of investigation include the visual observation of the process using high-speed photography,[1] the measurement of the pressure effects of the process,[1, 2] the sampling of the mixture during the process,[3] and spectral analysis of radiation emitted during the process.[4, 5] It has been established as a result of these investigations that the combustion process as it occurs in the engine cylinder consists of several reaction phases. During early studies of the combustion process, little work was done concerning the preflame reaction phase (the reaction which occurs in the charge ahead of the hot flame and before knock). In recent years, it has been proposed that preflame reactions and their products may play important roles in the phenomena of autoignition and knock. To date, however, the exact course of the chemical reactions which comprise the preflame reactions of any fuel-air mixture is not known.

In the light of this uncertainty, several methods other than a study of the exact chemical reactions involved have been applied to the problem. Two of these other methods which have proved of value are: (a) the evaluation of the net energy released during preflame reactions and (b) the observation of the radiation associated with preflame reactions. The net energy released as a result of preflame reactions can be evaluated from pressure-time data obtained from engine experiments. The radiation associated with preflame reactions can be observed by means of a photomultiplier tube as described by Pastell.[6]

In this paper an improved method for the evaluation of the net energy released by preflame reactions in engines is developed and compared with two previous methods. The improved method is applied to previous and new experimental data. Radiation measurements and net-energy-release results were obtained simultaneously in the new work for a variety of fuel-air mixtures. This enables a direct comparison to be made between these two separate effects of preflame reactions.

An Improved Method of Evaluating Energy Released During Preflame Reactions

A. *Thermodynamic Problem of Evaluating Preflame Energy Release* – The problem of evaluating the energy released during preflame reactions in an engine cylinder is essentially one of obtaining an energy balance for the fuel-air mixture in the cylinder. A simplified diagram of an engine is shown in Fig. 1, and the basic energy quantities involved are represented for the case in which a homogeneous reaction is taking place in the mixture. A balance of these energy quantities can be obtained for preflame reactions occurring in either a motored engine or a fired engine before the time of the spark. During the compression process while preflame reactions occur, the mixture is influenced by the following energy changes:

1. Work is done on the mixture by the piston.
2. Heat is exchanged between the mixture and the cylinder walls.
3. The internal energy of the mixture changes.
4. A net quantity of energy is added to or extracted from the mixture as a result of the preflame reactions occurring in the mixture. Net energy is defined as the algebraic sum of the

[1] See SAE Transactions, Vol. 33, (May) 1938, pp. 185-204: "Motion Pictures of Engine Flames Correlated with Pressure Cards," by G. M. Rassweiler and L. L. Withrow.

[2] See SAE *Quarterly* Transactions, Vol. 4, July, 1950, pp. 232-274: "Ignition of Fuels by Rapid Compression," by C. F. Taylor, E. S. Taylor, J. C. Livingood, W. A. Russell, and W. A. Leary.

[3] See Philosophical Transactions of Royal Society of London, Series A, Vol. 243, July, 19, 1951, pp. 463-524: "Study of Reactions that Lead to Knock in a Spark-Ignition Engine," by D. Downs, A. D. Walsh, and R. W. Wheeler.

[4] See *Industrial and Engineering Chemistry*, Vol. 27, August, 1935, pp. 872-879: "Effect of Tetraethyl Lead on Preflame Reactions in Engine," by L. L. Withrow and G. M. Rassweiler.

[5] See NACA Report No. 486 (1934), "Infrared Radiation from Explosions in Spark-Ignition Engine," by C. F. Marvin, Jr., F. R. Caldwell, and S. Steele.

[6] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 571-587: "Precombustion Reactions in a Motored Engine," by D. L. Pastell.

During Preflame Reactions

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This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1954.

energies of the exothermic and endothermic reactions which may occur.

The principle of the conservation of energy can be applied to the fuel-air mixture in terms of the four energy quantities listed above. A solution for the net energy released as a result of the preflame reactions over a given time period is then possible if the first three energy quantities can be evaluated over the same time period.

B. *Existing Methods of Evaluation of Net Energy Release* – One method which has been used for the evaluation of the energy effects of preflame reactions was originated by Rassweiler and Withrow.[1] This method was originally applied to the study of normal combustion in engines and later was adapted and used for evaluating preflame energy release by Cornelius and Caplan.[7] In this method of pressure record of the engine cycle during which preflame reactions occur is examined over small volume increments in the following manner. Referring to Fig. 2, the polytropic relationship (PV^n = constant) is used to calculate the pressure rise (Δp due to piston motion) over the volume increment chosen for the case of zero energy release. The difference between the observed pressure rise and the calculated portion due to piston motion for a given volume increment is the pressure rise due to energy release. Similarly, the pressure rise due to energy release can be evaluated for succeeding volume increments. The pressure rise due to energy release for each volume increment is adjusted to refer to some common cylinder volume by assuming the pressure rise is inversely proportional to the volume. These adjusted pressure rises for succeeding increments of volume can be plotted as a function of crank angle, and their summation can be considered a measure of the total net energy release. The actual process is, therefore, represented by a series of hypothetical incremental constant-volume processes.

[7] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 488-510: "Some Effects of Fuel Structure, Tetraethyl Lead, and Engine Deposits on Precombustion Reactions in a Firing Engine," by W. Cornelius and J. D. Caplan.

AN improved method, applicable to either motored or fired engines, for evaluating energy-release rate occurring during preflame reactions is described in this paper. A unique feature of this method is the use of a mechanical computer to handle calculations.

A smooth energy-release-rate curve can be obtained with this improved analysis technique – something not possible with two previous methods. However, the mean rate of energy release as determined by all three methods is in general agreement.

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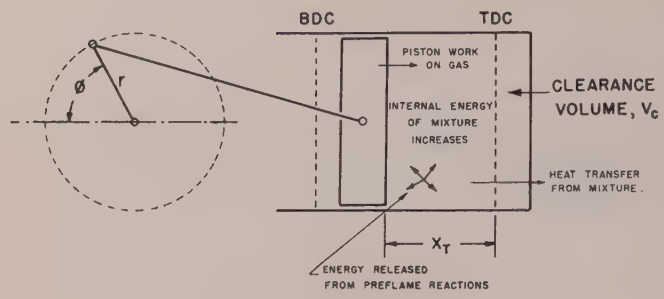

Fig. 1 – Sketch of engine mechanism

The method of Rassweiler and Withrow as applied to preflame reaction processes can be criticized on two points. (1) The method does not yield results directly in terms of the net energy released during preflame reactions. The pressure rise which is evaluated with the method is a secondary effect produced by preflame reactions. While this secondary effect may be useful in certain applications, the net energy released is a more basic result. (2) The method does not take into account all of the heat transferred from the mixture during preflame reactions; the results are therefore in error to some degree. This error arises because the actual heat transfer is greater during preflame reactions than the heat transfer accounted for by the polytropic exponent (n) determined before preflame reactions occur.

A second method which has been used to evaluate the energy released during preflame reactions was developed by Rifkin, Walcutt, and Betker.[8] In this method the various energy quantities shown in Fig. 1 are considered, and an energy balance is made for the fuel-air mixture. The net energy released is then evaluated from the energy balance. The heat transfer is evaluated from a separate test in which a fuel-air mixture which does not exhibit preflame reactions is used. It is then assumed that the total heat transfer is the same in the reacting as in the nonreacting case. The total net energy released during a preflame reaction process is obtained with a step-wise integration process.

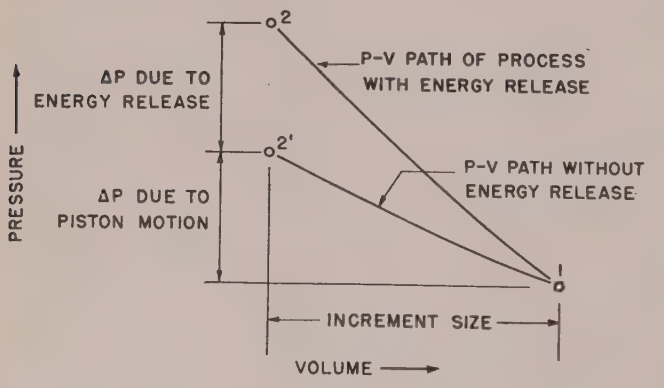

Fig. 2 – Incremental section of compression process with energy release

Although the method of Rifkin et al. is more tedious than that of Rassweiler and Withrow, it is more basic in its approach since it avoids the use of a hypothetical process. The method has, however, two shortcomings. First, the assumption that the heat transfer is the same in the reacting case as in the nonreacting case is in error. This was pointed out by Rifkin et al. who indicated that their method accounted for only a portion of the total heat transferred. Second, it is assumed that a fuel-air mixture can be found which does not exhibit preflame reactions but has the same thermodynamic properties as the fuel-air mixture under study. Since fuels which can be used for this purpose have unique thermodynamic properties, this requirement is difficult to fulfill exactly.

C. *An Improved Method of Evaluation* – The improved method for the evaluation of the net energy released during preflame reactions described here (see Appendix II) is based upon an energy balance for the fuel-air mixture in the engine cylinder. All the energy terms previously described and indicated in Fig. 1 are included. All of the heat transfer which occurs during the preflame reaction process from the mixture to the cylinder walls is included. This is accomplished by evaluating the heat transfer in terms of the temperature difference between the mixture and the engine coolant during the process and the conductance factor for the heat transfer between the mixture and the coolant. The conductance factor for heat transfer is determined from a separate test in which a fuel-air mixture which does not exhibit preflame reactions is used. The engine operating conditions used are similar to those used in the preflame reaction experiments. The final result of the analysis is represented by a single equation in terms of the engine operating conditions and pressure data from which the net *rate* of energy release may be evaluated at any point in the process. The final result is equation (14), Appendix II:

$$C_r^\circ = \frac{\pi \bar{c}_v}{MB_1(180)} \left\{ \frac{\bar{c}_{v0} M_0}{\bar{c}_v M} \left[P_0 k_0 \sin\phi - \left(\frac{dP}{d\phi}\right)_0 (C_1 + \cos\phi) \right] \left[\frac{P(C_1 + \cos\phi) - B_1 M T_c}{P_0(C_1 + \cos\phi) - B_1 M_0 T_c} \right] - \left[PK \sin\phi - \frac{dP}{d\phi}(C_1 + \cos\phi) \right] \right\}$$

where:

[8] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 472-487: "Early Combustion Reactions in Engine Operation," by E. B. Rifkin, C. Walcutt, and G. W. Betker, Jr.

C_r° = Net rate of energy release, Btu per mole of mixture per deg of crankshaft rotation

$\bar{c}_v$ = Mean value of constant volume specific heat for reacting mixture, Btu per mole per deg R

$\bar{c}_{v0}$ = Mean value of constant-volume specific heat for nonreacting mixture, Btu per mole per deg R

k = Ratio of mean specific heats for reacting mixture, $\bar{c}_p/\bar{c}_v$

k_0 = Ratio of mean specific heats for nonreacting mixture, $\bar{c}_{p0}/\bar{c}_{v0}$

M = Moles of mixture per cycle for reacting mixture, moles

M_0 = Moles of mixture per cycle for nonreacting mixture, moles

P = Cylinder pressure for reacting mixture, psf

P_0 = Cylinder pressure for nonreacting mixture, psf

$\frac{dP}{d\phi}$ = Rate of pressure rise for reacting mixture, psf per radians of crankshaft rotation

$\left(\frac{dP}{d\phi}\right)_0$ = Rate of pressure rise for nonreacting mixture, psf per radians of crankshaft rotation

B_1 = Engine constant, psf per deg R per mole

T_e = Engine coolant temperature, R

C_1 = Engine constant (dimensionless)

ϕ = Crankshaft position, radians, measured from bdc in the direction of the engine compression stroke

The nonreacting elements of equation (14) (subscript zero) are obtained from the separate test necessary to establish the heat-transfer conductance factor.

The assumptions used in the analysis (see appendix II) include the following: The total number of moles of mixture is constant during the preflame reaction phase, and the mixture can be described as a perfect gas.

The experimental data necessary to evaluate the net rate of energy release from equation (14) consist of the pressure-crank-angle data from the reacting and nonreacting experiments and the various engine operating constants. The engine operating constants can be classified as follows:

1. Mixture properties, $\bar{c}_v$, $\bar{c}_{v0}$, k, k_0.
2. Engine constants, B_1, C_1.
3. Engine operating conditions, M, M_0, T_e.

A typical set of pressure-crank-angle data obtained from a motored engine is shown in Fig. 3 and illustrates the required pressure data. An accurate pressure indicator is required for this procedure since pressure differences involved are relatively small. The pressure-crank-angle curve, as shown in Fig. 3, is obtained by drawing a smooth curve through the experimentally determined points. Four basic pressure terms, P, P_0, $dP/d\phi$, $(dP/d\phi)_0$, must be determined at each desired crank-angle position. These data are indicated in

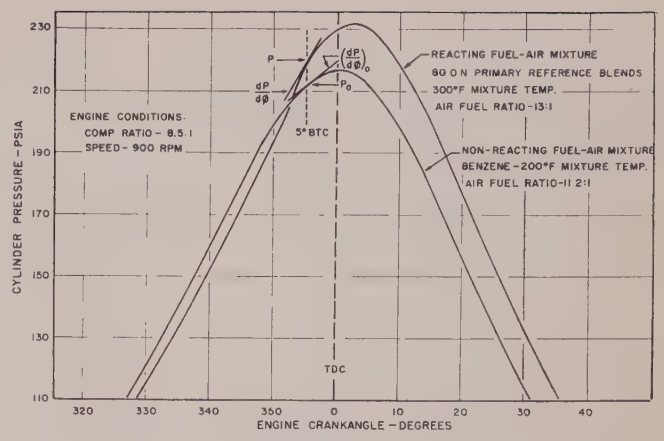

Fig. 3 – Basic pressure-crank-angle data from motored engine

Fig. 3 at a crank angle of 5 deg btc which is equivalent to $\phi = 175$ deg in equation (14).

All the calculations in this work including the slope calculation and the evaluation of equation (14) were done with an IBM computer where the input data fed to the computer were the pressures taken from the smoothed curves and the basic engine operating constants.

A typical net-energy-release-rate curve obtained with equation (14) is shown in Fig. 4. The pressure data illustrated in Fig. 3 were used, and a summary of the engine operating conditions is given in Table 1. The net energy liberated between any two crank-angle positions can be obtained by integration of the area under the curve shown in Fig. 4 between the two crank-angle positions; for example, in Fig. 4, between 340 deg btc and 13 deg atc, the total net energy release is approximately 1600 Btu per mole of charge.

The purpose of the nonreacting experiments is to reproduce the physical conditions which determine the conductance factor for heat transfer at the engine cylinder walls, such as the flow velocity pattern of the charge relative to the walls. The

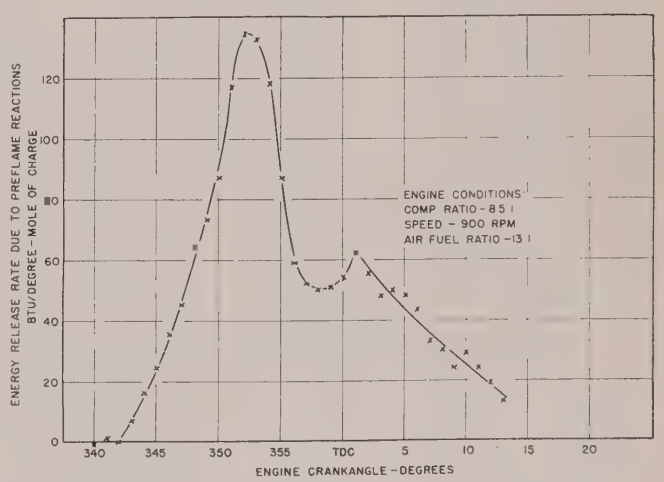

Fig. 4 – Energy-release-rate results for 80-octane primary reference blend evaluated from data shown in Fig. 3

Table 1 – Summary of Engine Operating Conditions

Engine	Cornelius-Caplan Custom-made single cylinder	Rifkin-Walcutt-Betker ASTM-CFR knock-rating	Authors ASTM-CFR knock-rating
Mode of Operation	Delayed ignition	Delayed ignition	Motored
Compression Ratio	6.7	5.62	8.5
Displacement	38.48	37.33	37.33
Coolant Temperature, F	175	205	212
Inlet-Mixture Temperature, F	300	212	Variable
A/F	13/1	12.7/1	13/1
Speed	800	900	900
Pressure Indicator	Balanced-pressure diaphragm	Balanced-pressure diaphragm	Balanced-pressure diaphragm

mixture temperature of the nonreacting mixture should not affect this conductance factor. If this is true, variation of the inlet-mixture temperature in the experiments with the nonreacting fuel should not cause any change in the energy release shown by a given reacting mixture. In Fig. 5 the results of an experiment are shown in which the test data of Fig. 3 for the reacting fuel were paired with each of five sets of nonreacting fuel data. The mixture temperature for the nonreacting fuel was varied from 150 F to 350 F in 50 F increments, and the results indicate that the inlet-mixture temperature of the nonreacting mixture does not influence greatly the net-energy-release rate obtained from a given reacting fuel-air mixture. Furthermore no trend is observed in Fig. 5 which would indicate that the benzene-air mixture is liberating net energy. The scatter of points at a given crank angle is attributed to the experimental error inherent in the data obtained for the various nonreacting fuel-air mixtures.

D. *Comparison of the Three Methods of Evaluation* – The results obtained with the two methods described in Section B which have been used previously for the evaluation of preflame energy release can be compared with the results given by the present method by applying all three methods to the same data. As described in Section B, the results normally obtained through the use of the method of Rassweiler and Withrow are not in terms of net-energy-release rate. It is therefore

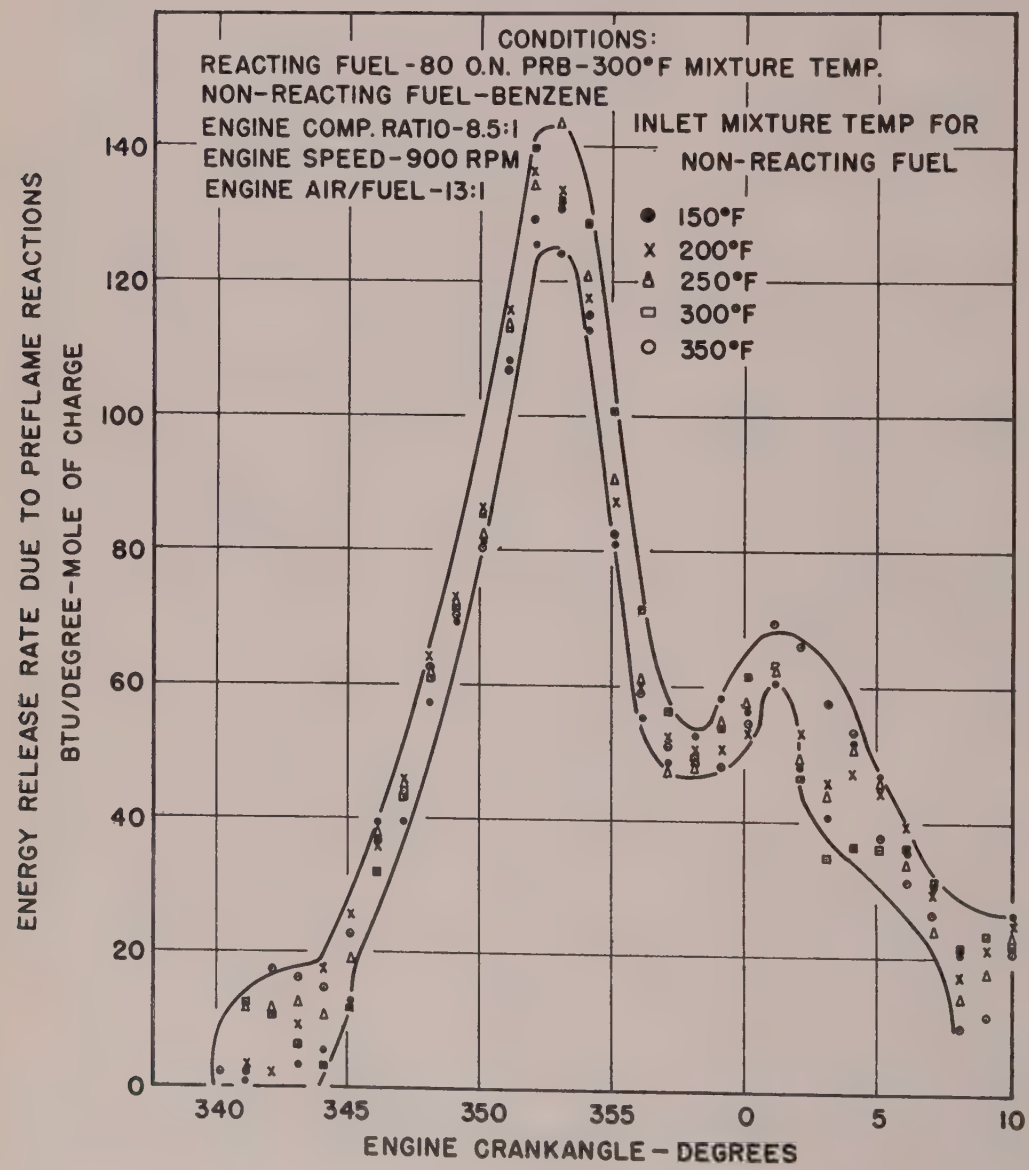

Fig. 5 – Energy-release-rate results for different nonreacting-fuel mixture temperatures

necessary to make additional assumptions to place their method on a comparable basis with the present method. The procedure used is outlined in Appendix III. The results obtained with the modified form of the method of Rassweiler and Withrow using the data shown in Fig. 3 are indicated in Fig. 6.

The method developed by Rifkin et al. was used by them mainly to evaluate the cumulative net energy released during preflame reactions. The method can be used directly, however, to evaluate the rate of net energy release over a selected increment of the process. The results thus obtained from the data shown in Fig. 3 are also indicated in Fig. 6.

The results shown in Fig. 4 for the net-energy-release rate are obtained from the data of Fig. 3 with the present method. The results are smoothed and shown replotted in Fig. 6 for direct comparison with the results of the other two methods.

The results obtained with the three evaluation methods, Fig. 6, indicate that there is general agreement between the three methods of evaluation. However, the scattering of points obtained with the two previously used methods may mask small effects. For instance, the second maximum rate of energy release as shown in Fig. 6 could not be identified with the results from these two previous methods of evaluation. The erratic points obtained with the two previously used methods are possibly due to the fact that in both methods the energy release must be evaluated over a selected finite increment of time and the energy released is then used to evaluate an average rate for the time interval involved. It is possible with the use of an optimum interval size and very accurate calculating procedures that the two previously used methods will give smooth results. The improved method enables an instantaneous rate of energy release to be evaluated at any time during the process.

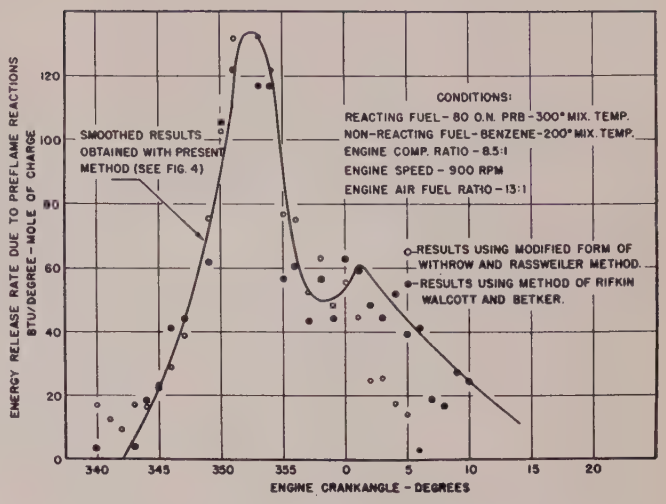
ENERGY RELEASE RATE DUE TO PREFLAME REACTIONS
BTU/DEGREE-MOLE OF CHARGE
SMOOTHED RESULTS OBTAINED WITH PRESENT METHOD (SEE FIG. 4)
CONDITIONS:
REACTING FUEL - 80 O.N. PRB - 300° MIX. TEMP.
NON-REACTING FUEL - BENZENE - 200° MIX. TEMP.
ENGINE COMP. RATIO - 8.5:1
ENGINE SPEED - 900 RPM
ENGINE AIR FUEL RATIO - 13:1
RESULTS USING MODIFIED FORM OF WITHROW AND RASSWEILER METHOD.
RESULTS USING METHOD OF RIFKIN WALCOTT AND BETKER.
ENGINE CRANKANGLE - DEGREES

Fig. 6 – Energy-release-rate results using three methods of evaluation

Results

While the primary objective of this paper is to introduce an improved method of evaluating energy release due to preflame reactions, it might be well to present some results obtained using the new method. The improved method of evaluation was used to analyze pressure data from three sources: (1) Rifkin, Walcutt, and Betker,[8] (2) Cornelius and Caplan,[7] and (3) the authors. Radiation traces were taken concurrently with the pressure data in the latter case to explore any possible correlation between net-energy-release rate and radiation from the reacting mixture.

Test Equipment and Engine Operating Conditions – The authors' experimental work was conducted with an ASTM-CFR knock-rating engine motored at a constant speed of 900 rpm with a compression ratio of 8.5/1. The engine was standard except for a new intake system. A balanced-diaphragm pressure indicator was placed in the spark-plug hole, and a quartz window was adapted to the bouncing pin hole. A cross-section of the combustion chamber is shown in Fig. 7. The regular CFR intake system was replaced with a carburetor and manifold better suited to the closed intake system employed. The carburetor was connected to the engine by 2 ft of 1½-in. diameter pipe. Two 1000-w heaters were used to elevate the mixture temperature. The mixture temperature was measured by means of a shielded thermocouple located in the inlet pipe close to the intake valve. Air to the carburetor was metered through critical flow orifices.[9] Fuel-flow rate was determined by timing the consumption of a known volume of fuel. Thus, the air/fuel ratio was determined from direct measurement of the air- and fuel-flow rates. The

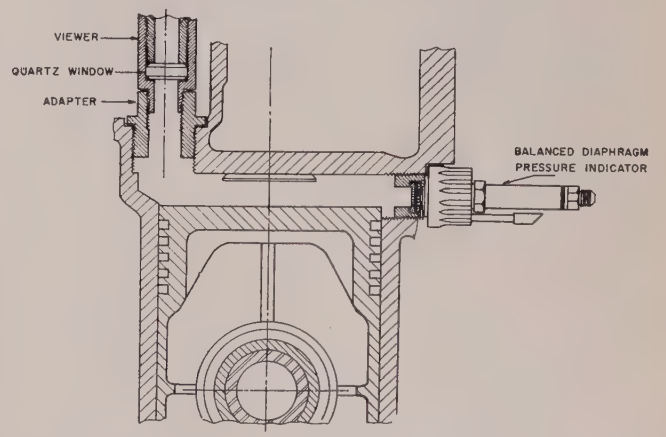

Fig. 7 – Cross-section of combustion chamber of CFR engine showing pressure-indicator and viewing-tube locations used by the authors

[9] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 666-676: "An Improved System for Control and Measurement of Air Consumption of a Single-Cylinder Engine," by W. Cornelius and J. D. Caplan.

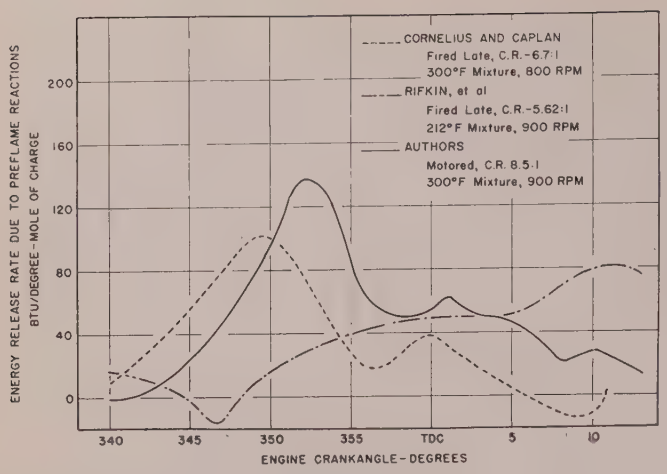

Fig. 8 – Energy-release rate of 80-octane primary reference blend in three engines

air/fuel ratio was approximately 85% of stoichiometric for all fuels. The number of moles of cylinder charge was determined in all cases by assuming a clearance-gas weight fraction of 10%. Water vapor in the intake air was less than 40 grains per pound of dry air for all tests. The pressure data for the nonreacting mixture was obtained with benzene as the fuel at an inlet-mixture temperature of 200 F.

The radiation detector used in the present work was an RCA 931A photomultiplier tube. The photomultiplier was positioned to receive radiation through a ¼-in.-thick, ⅜-in.-diameter quartz window which was sealed into the bouncing pin hole with a suitable holder. (See Fig. 7.)

Cornelius and Caplan[7] in their paper included pressure data with benzene as the fuel, an air/fuel ratio with 85.5% theoretical air, and an inlet-mixture temperature of 300 F. This pressure data is used for the nonreacting mixture data necessary for the improved method of evaluating energy release. Rifkin et al. did not include pressure data using benzene as the fuel; therefore, their pressure data for diisobutylene with an air/fuel ratio of

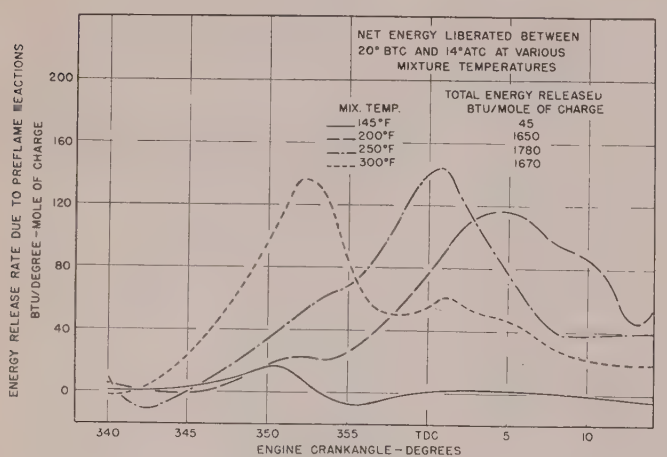
NET ENERGY LIBERATED BETWEEN 20° BTC AND 14°ATC AT VARIOUS MIXTURE TEMPERATURES
MIX. TEMP.
TOTAL ENERGY RELEASED BTU/MOLE OF CHARGE
145°F 45
200°F 1650
250°F 1780
300°F 1670
ENERGY RELEASE RATE DUE TO PREFLAME REACTIONS BTU/DEGREE-MOLE OF CHARGE
ENGINE CRANKANGLE-DEGREES

Fig. 9 – Energy-release rate of 80-octane primary reference blend at various inlet-mixture temperatures

12.7/1 and an inlet-mixture temperature of 212 F was used as the nonreacting mixture data. Other test conditions are summarized in Table 1.

Energy Release Results for Several Primary Reference Blends – In Fig. 8 are shown the results of the present method of evaluating energy release rate when applied to the pressure card data of three investigators. The 80-octane primary reference blend was a fuel common to the three investigations. The differences in the curves presumably reflect the effects of different compression ratios, mixture temperatures, modes of operation, and speeds. For instance, the Cornelius-Caplan curve (compression ratio 6.7, mixture temperature 300 F) shows preflame energy release commencing earlier than the Rifkin et al. curve (compression ratio 5.62, mixture temperature 212 F). The authors' curve (compression ratio 8.5, mixture temperature 300 F) probably lags behind the Cornelius-Caplan curve because the former reaction is in a motored engine with correspondingly lower surface temperatures. (The surface-temperature effect is evidently greater than the temperature effect due to the higher compression ratio.) One conclusion possible from a consideration of Fig. 8 is that engine operating conditions have a considerable effect on the energy-release time history.

In Fig. 9 are shown the energy-release-rate curves obtained by the authors for 80-octane primary reference blend at four mixture temperatures: 145 F, 200 F, 250 F, and 300 F. The results of the graphical integration of each curve are also given in Fig. 9. These integrations give the net positive energy released between 20 deg btc and 14 deg atc. It will be noted that the peak energy-release rate occurs earlier for higher mixture temperatures. One run was originally planned at a mixture temperature of 150 F; however, in carrying out the experiment, it was found that the net energy release and radiation were extremely variable from cycle to cycle. Further, it was found that lowering the mixture temperature to 145 F gave negligible cyclic variation and, as can be seen in Fig. 9, relatively little energy release. It appears from these data that 150 F, at the stated conditions of compression ratio, air/fuel ratio, and so forth, is the threshold mixture temperature above which the net energy release of 80-octane primary reference blend is fairly constant. Rifkin et al. reported a similar threshold effect of temperature on preflame energy release.

If the mixture temperature is held constant for a series of runs, in which only the reacting fuel is varied, and the rate curves are integrated over the range in which preflame reactions are evident, a plot such as Fig. 10 may be obtained. In this figure all points on a given line were obtained in the same engine under identical conditions, fuel being the only variable. The solid lines refer to primary ref-

erence blends of various octane number, and the broken lines refer to primary reference blends plus the stated amount of tel per gallon. For a given engine operating on primary reference blends there is apparently an approximately linear relationship between net energy released during preflame reactions and Research octane number; however, the line is displaced upwards if tel is added to the fuel. Also, the relationship between energy release and Research octane number depends on engine conditions since a change in engine conditions causes a displacement of the line. (Compare solid lines, Fig. 10.) A similar relationship dependent on octane number of primary reference blends was reported by Rifkin et al. and Cornelius-Caplan.[7]

Relation Between Energy Release Rate and Radiation Intensity – In Fig. 11 are shown the radiation traces with corresponding net-energy-release-rate diagrams for 80-octane primary reference blend at various inlet-mixture temperatures. Radiation amplification was constant except as noted. Engine conditions are listed in Table 1. As the inlet-mixture temperature was increased, the preflame energy release occurred earlier in the cycle as noted before. Similarly, the radiation trace commenced earlier with increasing mixture temperature. Furthermore, the major energy release coincided with an inflection or peak of the radiation trace. A second, minor energy-release-rate maximum corresponds to the peak intensity of the radiation. A possible explanation of this apparent anomaly would be that the first inflection point (or peak) on the radiation trace may correspond to the reaction occurring in the major volume of the combustion chamber, and the second larger radiation peak may consist of radiation from a reaction occurring in the volume created by the installation of the viewing tube (see Fig. 7) in addition to any radiation from the former reaction. The latter reaction occurs later in the cycle perhaps due either to a lower wall temperature or to the increased surface-to-volume ratio in the viewer cavity. In any event, this later reaction could possess a greater radiating depth with a much smaller volume than the earlier reaction; hence, the radiation flux would be greater, and the associated energy release would tend to be minor. However, the relationship becomes indistinct at the lower mixture temperatures.

In Fig. 12 are shown the corresponding rate diagrams and radiation traces for primary reference blends of 80, 85, 90, 95, and 100 octane, all at a mixture temperature of 300 F. The correlation between energy-release rate and radiation intensity for this series can also be explained by the above mechanism, except for the higher octane number blends, in which the separation of the two peaks becomes indistinct.

The radiation and energy-release-rate curves obtained when 3 ml of tetraethyl lead per gallon

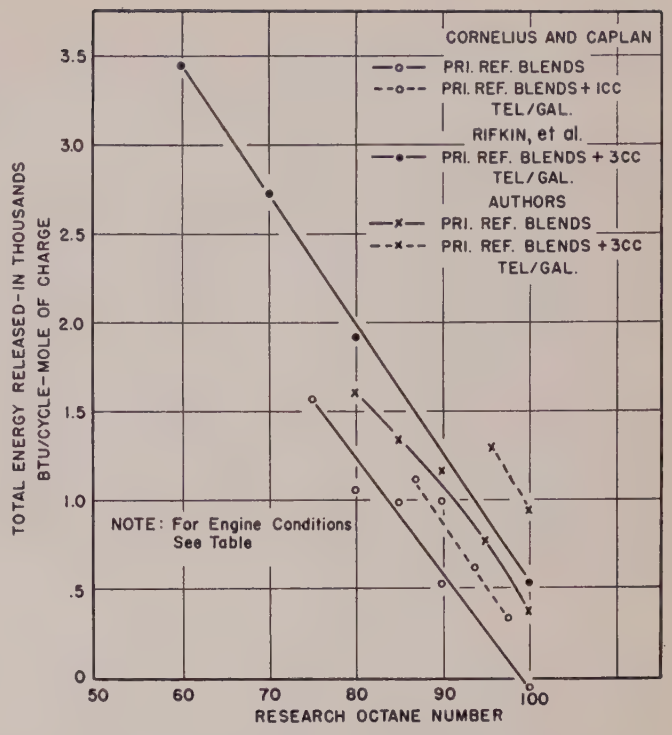

Fig. 10 – Net energy released due to preflame reactions as a function of Research octane number for the various sources

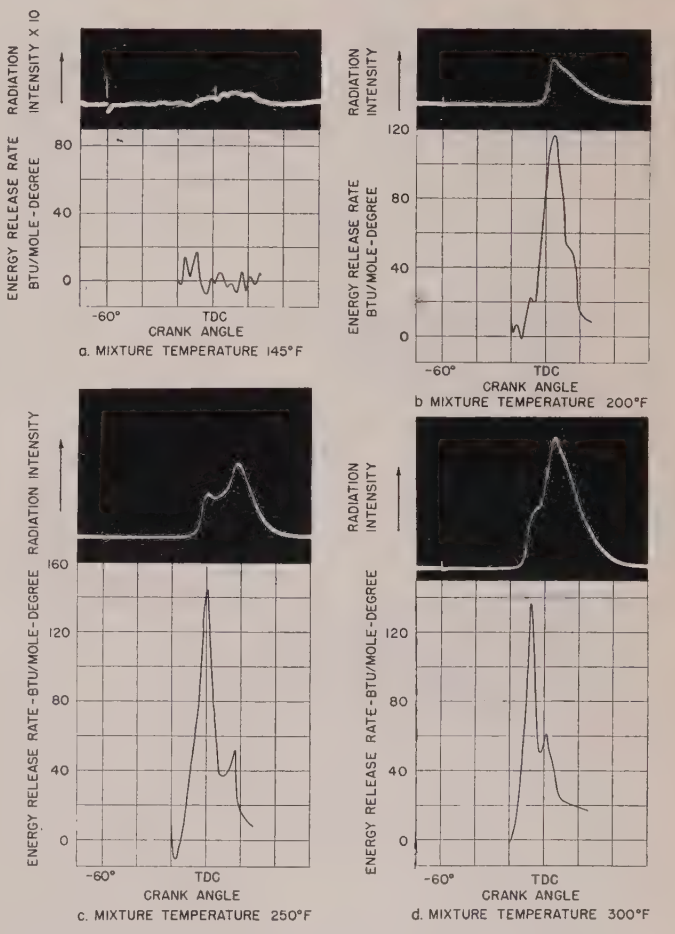

Fig. 11 – Energy-release rate and radiation traces for 80-octane primary reference blend with various inlet-mixture temperatures

of fuel are added to 80- and 90-octane primary reference blends and isooctane are shown in Fig. 13. Under these conditions the correlation between energy-release rate and radiation is not as good as has been shown with the unleaded primary reference blends. By comparing the results obtained with the leaded fuels shown in Fig. 13 with the corresponding results obtained with the unleaded fuels shown in Fig. 12, it is found that both the maximum radiation intensity and the maximum energy-release rate are decreased by the addition of 3 ml of tetraethyl lead per gallon of fuel. In addition by comparing Figs. 12e and 13a, it is found that the ratio of the second peak to the first peak in the radiation trace is decreased when the lead is added to 80-octane primary reference blend and that the ratio of the second peak to the first peak in the energy-release-rate curve is increased when the lead is added.

Summary

1. An improved method has been developed for the evaluation of the energy-release rate which occurs during preflame reactions in engines. The method can be applied to either motored or fired engines and is valid in the latter case up to the point in the engine cycle when the spark is fired.

2. It has been possible with the improved method of analysis to obtain a smooth energy-release-rate curve for preflame reactions in engines. Two previous methods of analysis do not show this detailed effect; however, the mean rate of energy release as determined with the three methods is in general agreement.

3. Since the improved method requires data from a nonreacting mixture test to evaluate the heat-transfer effects of the engine, various inlet-mixture temperatures of the nonreacting fuel were combined with a given set of reacting mixture data to determine the effect upon the energy-release results. Although some variation of the energy-release time history is evident from these data, the essential features of the energy-release-rate curve are retained.

4. The total energy released per mole of charge as a result of preflame reactions is dependent both on the fuel and the engine conditions. This confirms the findings of previous investigators.

5. It was found that for one fuel-air mixture the total energy released per mole of charge as a result of preflame reactions is relatively independent of the inlet-mixture temperature. This confirms the hypothesis of Rifkin et al.

6. A correlation was found to exist between the energy-release rate and the radiation effects of preflame reactions.

Acknowledgment

The writers take this opportunity to express their appreciation to Dr. Lloyd Withrow, John Caplan, and Walter Cornelius for their aid and

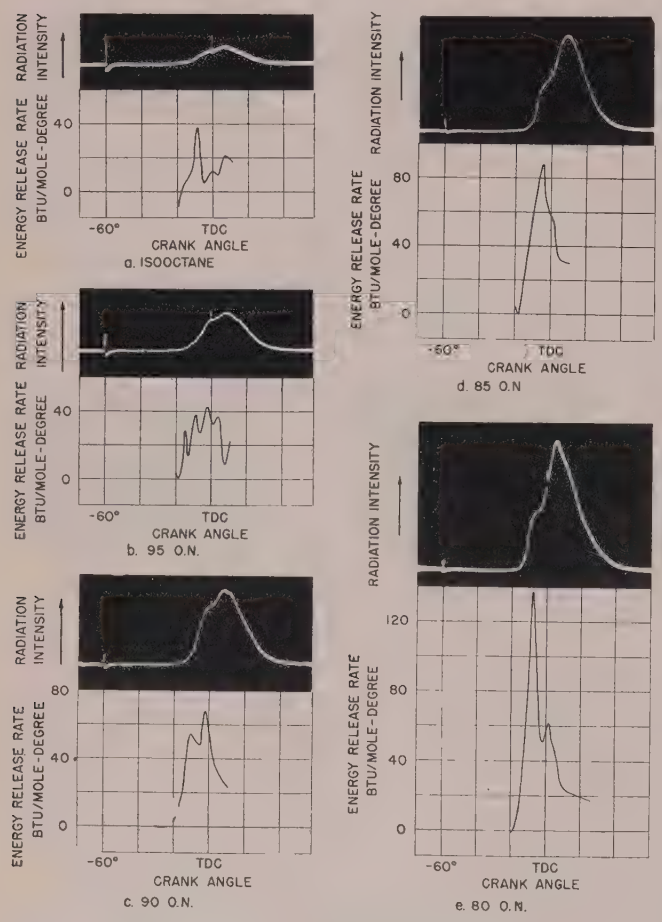

Fig. 12 – Energy-release rate and radiation traces for various primary reference blend fuels motored with a mixture temperature of 300 F

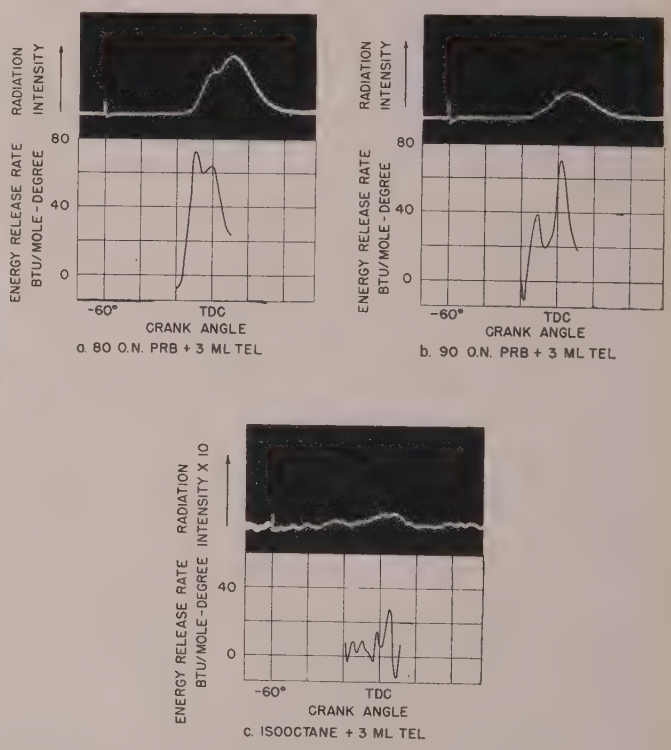
ENERGY RELEASE RATE BTU/MOLE-DEGREE
RADIATION INTENSITY
RADIATION INTENSITY X 10
-60°
TDC
CRANK ANGLE
a. 80 O.N. PRB + 3 ML TEL
b. 90 O.N. PRB + 3 ML TEL
c. ISOOCTANE + 3 ML TEL

Fig. 13 – Energy-release rate and radiation traces for various primary reference fuels with tel and an inlet-mixture temperature of 300 F

understanding in this work and to Ellis Rifkin, Cleveland Walcutt, and G. W. Betker of the Ethyl Corp. for their kind cooperation in supplying copies of the original data used for their publication.

APPENDIX I

The definitions of symbols are:

A = Heat-transfer area, sq ft

B_1 = Engine constant, $\frac{\text{Universal gas constant}}{\frac{1}{2}\text{ engine displacement}}$, lb per sq ft per mole per deg R

b = Engine bore, ft

C = Engine compression ratio

$C_1 = \frac{C+1}{C-1}$

C_r = Instantaneous rate of net energy release, Btu per radian of crankshaft rotation

C_r° = Instantaneous rate of net energy release, Btu per degree of crankshaft rotation per mole of mixture

$\bar{c}_v$ = Mean value of constant-volume specific heat for reacting mixture, Btu per mole per deg R

$\bar{c}_{v0}$ = Mean value of constant-volume specific heat for nonreacting mixture, Btu per mole per deg R

D = Heat-transfer parameter, $U_0A/2\pi N$, Btu per deg R

E_r = Chemical energy released by the (assumed) homogeneous fuel-air mixture as a result of preflame reactions, Btu

k = Ratio of mean specific heats for reacting mixture, $\bar{c}_p/\bar{c}_v$

k_0 = Ratio of mean specific heats for nonreacting mixture, $\bar{c}_{p0}/\bar{c}_{v0}$

M = Moles of fuel-air mixture in the cylinder (considered constant)

M_0 = Moles of nonreacting fuel-air mixture in the cylinder

n = Polytropic exponent

N = Engine speed, rpm

P = Pressure of reacting fuel-air mixture, psf

P_0 = Pressure of nonreacting fuel-air mixture, psf

Q = Heat transferred, Btu

R = Universal gas constant, 1.986 Btu per mole per deg R

r = One-half of engine stroke, ft

T = Temperature of fuel-air mixture, R

T_c = Temperature of engine coolant, R

U_0 = Overall heat-transfer coefficient from fuel-air mixture to engine coolant, Btu per min per sq ft per deg R

U = Internal energy of fuel-air mixture, Btu per mole

V = Cylinder volume, cu ft

V_c = Clearance volume, cu ft

W_{in} = Work done by the piston, Btu

X_t = Piston travel from tdc, ft

$(dP/d\phi)$ = Slope of pressure-crank-angle curve for reacting mixture, lb per sq ft per radian

$(dP/d\phi)_0$ = Slope of pressure-crank-angle curve for nonreacting mixture, lb per sq ft per radian

θ = Time, min

ϕ = Crank angle measured from bdc, radians

δ = Denotes size comparable to differential, but does not identify variable as an exact differential.

APPENDIX II

Analysis for Energy Release Rate Due to Preflame Reactions – The preflame reaction phase of the combustion process as it occurs in the engine can be examined in detail during a compression and expansion process in the engine in the absence of a hot flame. The preflame chemical reactions which may occur within the fuel-air mixture can be either exothermic or endothermic in nature. In any case, the net effect of these reactions from an energy point of view will be either to add energy to or absorb energy from the mixture. The purpose of this analysis is to evaluate the net energy released as a result of preflame reactions.

During the compression and expansion processes, the mixture in the cylinder is subjected to various energy transformations. The energy changes involved are as follows:

1. The piston does work on the mixture during compression.

2. The mixture does work on the piston during expansion.

3. Heat transfer occurs between the mixture and the cylinder walls.

4. The internal energy of the mixture changes.

5. A net quantity of energy is added to or extracted from the mixture as a result of preflame reactions.

The energy balance for the mixture may then be expressed as

Piston work in + Net energy released = Heat transfer out + Increase of internal energy of mixture.

A sketch of the cylinder with the various energy forms indicated is shown in Fig. 1.

The following assumptions are made throughout the analysis:

1. It is assumed that the unreacted fuel-air mixture and any subsequent reaction products comprise a homogeneous mixture throughout the physical process described above and that this mixture can be described as a perfect gas.

2. It is assumed that the preflame reactions occur in the engine cylinder before the spark initiates a so-called "hot flame" in the fuel-air mixture. Thus, the analysis is intended to apply to the situation where a retarded spark is used in a fired engine and the preflame-reaction energy release is studied in any portion of the process before the spark. Obviously the analysis also applies to motored engines.

3. It is assumed that the number of moles of reacting mixture remains constant during the preflame reactions and that a constant mean value of specific heat with respect to both composition and temperature can be used to evaluate the heat capacity of the mixture.

The energy balance for the mixture described previously may be written as:

$$\delta W_{in} + \delta E_r = \delta Q_{out} + M dU \qquad (1)$$

where the equation is written for the quantity of mixture in the engine cylinder and the significance of the various symbols is defined in Appendix I. Since the mixture is assumed to remain in the gas phase,

$$dU = \bar{c}_v dT \tag{2}$$

Also, the work term may be written as:

$$W_{in} = -PdV/778 \tag{3}$$

Since heat transfer is a path-type function, the δQ_{out} term as it appears in equation (1) must be evaluated in terms of the phenomena which control its magnitude. For this case it will be assumed that the instantaneous heat-transfer rate from the mixture to the cylinder walls can be approximated by using the engine coolant temperature for the wall temperature. While this is not strictly true, it is approached at the time of the application of the method. The instantaneous rate of heat transfer between the mixture and the engine coolant is dependent upon a combined effect of convection and conduction. The δQ_{out} which occurs in time $d\theta$ can be expressed as:

$$\delta Q_{out} = U_0 A(T - T_c)d\theta \tag{4}$$

in which all the terms with the exception of T_c will be functions of time.

The second term of equation (1), the net energy released as a result of preflame reactions, can be expressed in terms of the instantaneous net rate of energy release exhibited by the mixture. Thus:

$$\delta E_r = C_r d\phi \tag{5}$$

in which C_r will be an unknown function of time.

Substituting equations (2), (3), (4), and (5) into equation (1):

$$M\bar{c}_v dT + U_0 A(T - T_c)d\theta + \frac{PdV}{778} - C_r d\phi = 0 \tag{1a}$$

The perfect-gas equation can be written in the following forms:

$$PV/R778 = MT \tag{6}$$

$$MdT = \frac{PdV + VdP}{R(778)} \tag{6a}$$

Substituting equation (6a) into (1a) gives:

$$\frac{\bar{c}_v}{R}\frac{(PdV + VdP)}{778} + U_0 A(T - T_c)d\theta + \frac{PdV}{778} - C_r d\phi = 0 \tag{1b}$$

The essential features of the engine mechanism involved in the process are shown in Fig. 1. The volume in the cylinder at any piston position can be expressed as:

$$V = V_c + \frac{\pi}{4} b^2 X_t \tag{7}$$

in which the clearance volume, V_c, can be expressed as:

$$V_c = \frac{\pi r b^2}{2(C - 1)} \tag{8}$$

in which C is the compression ratio of the engine.

With the assumption that the piston moves with simple harmonic motion (independent of connecting-rod length), X_t can be expressed as:

$$X_t = r(1 + \cos\phi) \tag{9}$$

where the crank angle ϕ is measured from bdc as indicated in Fig. 12. Substituting equations (8) and (9) into equation (7) gives:

$$V = \frac{\pi}{4} b^2 r\left[\left(\frac{C + 1}{C - 1}\right) + \cos\phi\right] \tag{7a}$$

and

$$dV = -\frac{\pi}{4} b^2 r \sin\phi\, d\phi \tag{10}$$

Combining equations (6) and (7a) gives:

$$T = \frac{P\pi b^2 r}{MR(778)(4)}(C_1 + \cos\phi) \tag{11}$$

in which

$$C_1 = \frac{C + 1}{C - 1}$$

Substituting equations (7a), (10), and (11) into equation (1b) and simplifying gives:

$$\frac{dP}{d\phi} - \frac{Pk\sin\phi}{(C_1 + \cos\phi)} + \frac{PD}{M\bar{c}_v} - \frac{B_1 T_c D}{\bar{c}_v(C_1 + \cos\phi)} - \frac{B_1 C_r}{\bar{c}_v(C_1 + \cos\phi)} = 0 \tag{12}$$

in which

$$D = \frac{U_0 A}{2\pi N} \quad \text{and} \quad B_1 = \frac{(4)(778)R}{\pi b^2 r}$$

Upon examination of equation (12) it becomes clear that the variable heat-transfer parameter, D, which includes the concepts of conductance and area, must be determined before a solution for C_r can be obtained in terms of the engine pressure-crank-angle data. One method which can be used to evaluate the variable heat-transfer parameter consists of operating the engine in such a way as to duplicate the heat-transfer characteristics in the engine while the net-energy-release rate is zero. This can be accomplished by operating the engine with a mixture which is nonreacting in the portion of the engine cycle under study. Inasmuch as the engine speed is perhaps the most important element of the heat-transfer parameter, D, it is essential that the speed in the nonreacting case be the same as in the reacting case. Obviously, the combustion-chamber configuration, valve timing, inlet valve, and the induction and exhaust system must be the same in the nonreacting case as in the reacting case.

Applying equation 12 to the nonreacting case ($C_r = 0$) and solving for D gives:

$$D = \frac{\left(\frac{dP}{d\phi}\right)_0 - \frac{P_0 k_0 \sin\phi}{(C_1 + \cos\phi)}}{\frac{B_1 M_0 T_c}{M_0 \bar{c}_{v0}(C_1 + \cos\phi)} - \frac{P_0}{M_0 \bar{c}_{v0}}} \tag{13}$$

in which the zero subscript notation denotes the nonreacting case. Resubstituting equation (13) back into equation (12) and solving for C_r gives:

$$C_r = \frac{\text{Btu}}{\text{radian}} = \frac{\bar{c}_v}{B_1}\left\{\frac{\bar{c}_{v0}M_0}{\bar{c}_v M}\left[\frac{P(C_1+\cos\phi) - B_1 M T_c}{P(C_1+\cos\phi) - B_1 M_0 T_c}\right]\right.$$
$$\left[(P_0 k_0 \sin\phi) - \left(\frac{dP}{d\phi}\right)_0 (C_1 + \cos\phi)\right]$$
$$\left. - \left[Pk\sin\phi - dP\,(C_1+\cos\phi)\right]\right\} \tag{14a}$$

or

$$C_r^\circ = \frac{\text{Btu}}{(\text{degree})(\text{mole})}$$
$$= \frac{\bar{c}_v \pi}{(180) M B_1}\left\{\left[\frac{c_{v0}M_0}{\bar{c}_v M}\,\frac{P(C_1+\cos\phi) - B_1 M T_c}{P_0(C_1+\cos\phi) - B_1 M_0 T_c}\right]\right.$$
$$\left[(P_0 k_0 \sin\phi) - \left(\frac{dP}{d\phi}\right)_0 (C_1 + \cos\phi)\right]$$
$$\left. - \left[(PK\sin\phi) - \frac{dP}{d\phi}(C_1+\cos\phi)\right]\right\} \tag{14b}$$

The result shown in equation (14) represents the final form obtained with this analysis for the evaluation of the instantaneous net-energy-release rate due to preflame reactions in an engine cylinder. The data needed to evaluate C_r as a function of crank angle consist mainly of the pressure-crank-angle results of engine tests for both the nonreacting mixture (in the portion of the engine cycle under consideration) and the reacting mixture.

APPENDIX III

Modification of the Method of Rassweiler and Withrow – The method of Rassweiler and Withrow for the evaluation of the pressure development due to chemical reactions in an engine does not furnish results which are directly comparable with the results given by the present improved method. It is therefore necessary to make additional assumptions to produce a result which is comparable. The following approximations were used: In Fig. 2, the pressure rise due to combustion as defined by Rassweiler and Withrow, apart from that due to piston motion, was assumed to have been caused by an incremental, constant-volume heating process. Thus the incremental energy necessary to cause the temperature rise from T'_2 to T_2 at constant volume was assumed to equal the net energy release due to the precombustion reactions during the time interval involved. So, the net energy required for this hypothetical constant-volume incremental process becomes:

$$_1Q_2 = Mc_v(T_2 - T'_2) \tag{15}$$

Since the process is constant volume:

$$T_2 = T'_2 \frac{P_2}{P'_2} \tag{16}$$

Since the process from 1 to 2′ is assumed to be a polytropic process with the value for n assumed constant:

$$T'_2 = T_1\left(\frac{V_1}{V_2}\right)^{n-1} \tag{17}$$

and

$$P'_2 = P_1\left(\frac{V_1}{V_2}\right)^n \tag{18}$$

The perfect-gas law applied in condition 1 gives:

$$T_1 = \frac{P_1 V_1}{MR(778)} \tag{19}$$

Substituting equations (16), (17), (18), and (19) into equation (15) gives:

$$_1Q_2 = \frac{c_v P_1 V_2}{R(778)}\left[\frac{P_2}{P_1} - \left(\frac{V_1}{V_2}\right)^n\right] \tag{20}$$

Thus, equation (20) gives the energy release required for the assumed fictitious constant-volume process for the increment. Obviously the accuracy of the approximation improves as the increment size decreases. The value of the polytropic exponent can be obtained directly from the pressure-volume data for the reacting mixture before the effects of precombustion reaction become evident.

DISCUSSION

Believe Density of Charge Does Affect Conductance

– O. A. Uyehara and P. S. Myers
University of Wisconsin

THE study of preflame reactions has been of considerable interest during the past years, and the authors are to be congratulated for adding to our store of information concerning the extent of these reactions. It is hoped that studies using techniques suggested in the paper will eventually assist us in understanding the complex combustion phenomena in the engine.

Since the authors present primarily a technique rather than extensive data on preflame reactions, let us look at their technique. It differs from previous techniques in at least two ways: the use of a computing machine and the method of evaluating the heat transferred to or from the cylinder.

The use of a computing machine certainly enables laborious calculations to be made that otherwise would not be attempted. The fitting of the curve with an equation and obtaining the slope by differentiating the equation is especially good, for obtaining slopes by graphical differentiation is notoriously inaccurate. We wonder, however, why the curve was first hand drawn and then an equation fitted to this hand-drawn curve. Would it not have been better to

determine the curve by a least-squares technique rather than by drawing the curve in by hand (especially when it was already planned to use the computing machine), or were the data good enough that only a single curve could be drawn through the data?

The other way in which the authors' technique differed from those used previously was in the method of evaluating the heat transfer to and from the mixture. This is presumably accomplished by determining the value of the conductance in a nonreacting cycle. This is certainly a basic approach and, as such, is good. However, is the assumption that a nonreacting fuel can be found having the same conductance as the fuel in question any better than the assumption that a nonreacting fuel can be found having the same thermodynamic properties as the fuel in question?

And speaking of conductance, we would like to question the authors' explanation of the results shown in Fig. 5. The statement is made in connection with Fig. 5 that varying the mixture temperature of the nonreacting mixture should not vary the conductance. However, varying the mixture temperature varies the density of the charge. Fig. A shows a similar situation where the density of the charge was varied and the heat transfer did increase. Since the mixture temperature was not changed, we must conclude that the density does affect the conductance in the internal-combustion engine. Would not a more plausible explanation of Fig. 5 of the paper be that as the mixture temperature was increased the temperature of the charge was increased and the conductance decreased to produce the effect noted?

Since the authors (as well as other investigators) have based their evaluation of the energy released in the preflame reactions on a reference nonreacting cycle, would it not seem logical to study this nonreacting cycle in some detail? There is perhaps a little malice or frustration in this thought, for we have attempted to analyze such reference cycles with not too much success. Fig. B shows one of the more successful trials. The data were obtained during a single hot cycle on a GM I-71 diesel engine using a Li indicator and a balanced-diaphragm indicator for calibration. Every attempt was made to secure an accurate diagram. The data were then plotted on a log pressure versus log volume diagram and analyzed in an attempt to determine the amount of heat transfer. As is shown in Fig. B, the amount of heat transferred was appreciable with respect to the energy released in preflame reactions.

In conclusion, we would like to inquire if the error in assuming simple harmonic motion was negligible and to express the hope that the technique will be used to obtain more information on precombustion reactions.

Presence of Cool Flame May Upset Determinations

– E. B. Rifkin
Ethyl Corp.

THE authors have made a valuable contribution to the thermodynamics of precombustion reactions. Perhaps their major advance has been in the adaptation of previously developed methods to machine calculation. It is hoped that recognition of the fact that fundamental investigations into behavior of fuels in engines can be made simpler and less tedious by this technique will encourage others to enter this promising field. The second improvement introduced in this work, namely, that of a more rigorous treatment of heat transfer, is also to be highly commended because it does minimize an important area of uncertainty.

The most difficult aspect of all treatments of heat effects due to precombustion reactions in engines is the problem of evaluating these effects in the presence of a normal flame. In conventional applications, it is the preflame reactions which occur ahead of the flame front which are the most important and which must be studied carefully before a complete understanding of the knocking mechanism can be had. Although some previous precombustion studies made in the laboratories of Ethyl Corp. have been carried out in the presence of a flame front, the mathematical methods become quite uncertain because the thermodynamic procedures can only apply to a homogeneous system and it is necessary to know just where, in terms of mass and volume, the flame front is at any time.

Although it has not been generally recognized, the same question can also arise in the case of the motored engine. Recent experiments in our laboratories have shown that under certain conditions it is possible to see a cool flame traverse the combustion chamber of an L-head engine reproducibly cycle after cycle, in much the same way as a normal flame can traverse the chamber. Since numerous investigators, including the present authors, have shown that cool flames are closely related to the energy liberated by precombustion reactions, this raises the question as to whether the motored engine data which show appreciable heats of precombustion reactions also involve a cool flame traversing the chamber. If they do, which seems quite likely, then the treatment of these data by the present authors, as well as the two groups of earlier workers, must be further investigated to show that only homogeneous

DISCUSSION CONTINUED ON PAGE 650

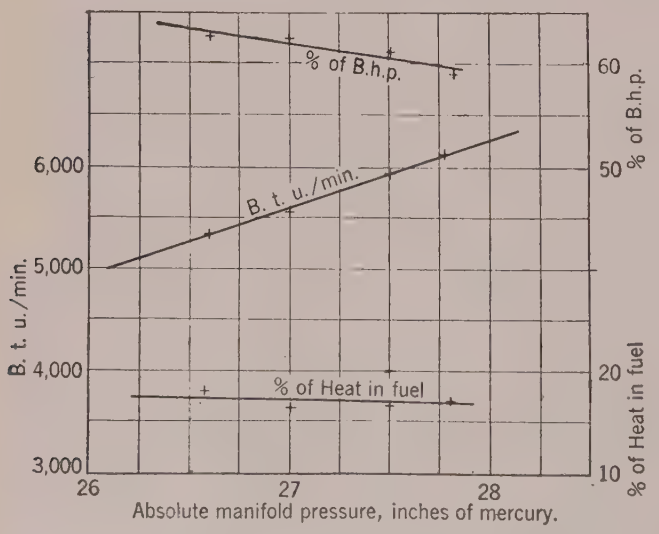

Fig. A – Effect of inlet pressure, constant rpm, on heat rejected to cooling medium of liquid-cooled engines

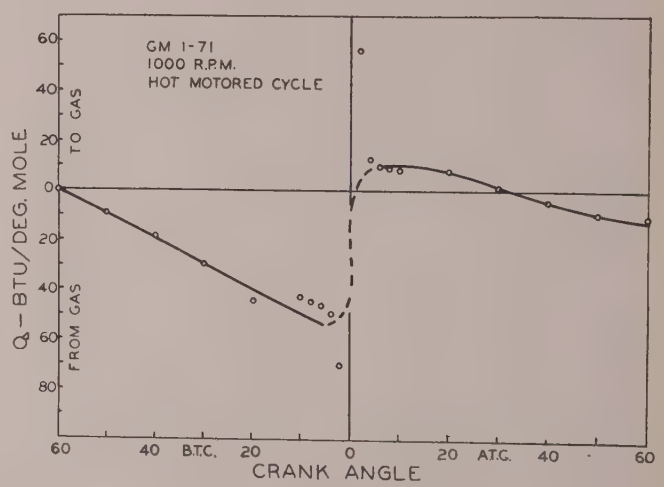

Fig. B – Energy-release rate of GM 1-71 diesel engine during hot motored cycle

A Review of Some Contemporary British Passenger-Car Engines

C. L. Goodacre, Clifford Motor Components, Ltd.

This paper was presented at the SAE National Passenger-Car, Body, and Materials Meeting, Detroit, March 3, 1954.

INFORMATION on some of the more popular and interesting British passenger-car-engine designs, which are in quantity production today, is presented here. It may be noted that there is some conflict of opinion in British engine design. This would appear to be a healthy sign in a thriving industry. The author has tried to present designers' reasons as to why they did certain things in the design of their engines.

Some engines reviewed are of prewar design when the United Kingdom suffered under an automobile horsepower taxation formula that was based on total piston area, the formula being the diameter of the cylinder squared (in inches) x the number of cylinders x a factor of 2.5. It is known as the Royal Automobile Club formula.

Many British engine designers considered this formula a millstone around their necks in that they were almost forced into small-bore, long-stroke engines to keep down the tax on the car. Because of pressure from the industry in the postwar period, a tax based on this formula was abandoned in 1947 in favor of a flat tax on all passenger cars irrespective of size, which now stands at £12 yearly (approximately $34).

It will be noted that there is a definite tendency in Britain today to design engines with cylinder proportions approaching square and in some cases over-square, although it must be pointed out that some engines that have quite a long stroke and thus, a high mean piston speed, function very satisfactorily and show good resistance to wear and increase in oil consumption. Some of these latter engines are in the high-performance bracket, that is, over 40 bhp for 60 cu in. of cylinder capacity.

For the British engine designer, one of the main factors for consideration with the more popular low-priced car is that of fuel economy. British designers were also very limited by fuel quality between 1939 and 1953, which, in the U. K. and many export markets that were open to British producers, was a single grade of gasoline of 70 octane number or less (Motor Method), with very low sensitivity. Premium fuels, which were available in the U. K. and elsewhere before the war,

DESIGN details for many of the passenger-car engines being built in Britain are presented here. Illustrations are used, wherever available, to make the details clear to the reader.

The author points out that the well-known Royal Automobile Club formula is no longer used in determining the British vehicle tax. This formula was considered by many British designers a millstone around their necks. It is based on piston area, and so they were almost forced to design their engines with small bore and long stroke to keep down taxes on the car.

Thus, the author reveals, the present tendency seems to be toward cylinder proportions approaching square and, in some cases, even over-square.

The Author

C. L. GOODACRE (M '46) is Technical Manager of Clifford Motor Components, Ltd., Birmingham, England. During World War II, he was assigned to the Directorate of Technical Development in the Ministry of Aircraft Production. Mr. Goodacre was with the Associated Ethyl Co., Ltd., of London before joining Clifford.

have only been reintroduced in the U. K. since February, 1953, and now there are premium fuels available in Britain today of 82 octane number (Motor Method), 94 octane number (Research). Fortunately, such better fuels are also becoming available in more of the export markets as time goes on. Many postwar engine designs are such that advantage can be taken of these better fuels with minor modifications.

However, the engine designer for the lower-priced car has to consider the increase in cost of the better fuels, which is in the order of 6-8% over the regular fuel, which in the U. K. is running around 72 octane (Motor Method), 76 octane (Research). It is a moot point when striking a balance of compression ratio as to which fuel to design for. It usually occurs that the cheaper cars are designed to take the maximum out of the regular fuel and can partially appreciate the premium fuel by an increase in spark advance.

The more expensive cars now appear to be about 80% of the way toward premium fuel and they can accept regular fuel with some spark retard, mild detonation, and drop in engine performance.

The author thinks that the British motor industry, among its many economic handicaps in the postwar era, from an engine point of view has suffered considerably from fiscal restrictions such as the piston area formula and the tax that is levied on the car and on its fuel – the regular gasoline costing 54¢ per U. S. gallon at present rates, of which 60% is tax.

In producing new engines in the postwar period, there appears the universal problem of heavy increase in tooling costs since prewar days. This in itself seems to dictate that any new design shall be such, and so proved, that it will have around 15 years effective life as a basic engine for quantity production. This fact does make for better engines from the onset.

From the author's knowledge of the industry, present engines reviewed in this paper, while they are contemporary British passenger-car engines, are in many cases considered obsolescent by their builders. These manufacturers are always pressing on with new designs, which seem, at this day and age, to take some three to five years to evolve into production form from the initial conception.

Alvis 6-Cyl Engine

The Alvis 3-liter, 6-cyl engine (Figs. 1 and 2) is manufactured by Alvis Ltd., of Coventry.

In deciding the type of engine for a medium-sized passenger car of 3200 lb gross weight, the Alvis designers chose a fairly orthodox 6-cyl in-line layout of 182.6 cu in. capacity.

This engine was designed from a clean sheet of paper in the immediate postwar period without paying attention to "tax engineering." It was felt that this layout was generally a more rugged job than a V-type unit or other alternative cylinder arrangements, bearing in mind the size of vehicle the company wished to build and the acceptability of that vehicle by the customer.

In the opinion of the Alvis designer, the weaknesses of this type of engine are fairly well known, and where a weakness is known, intelligent design can usually minimize this weakness so that it is practically acceptable. The long crankshaft with this type of engine may be considered undesirable

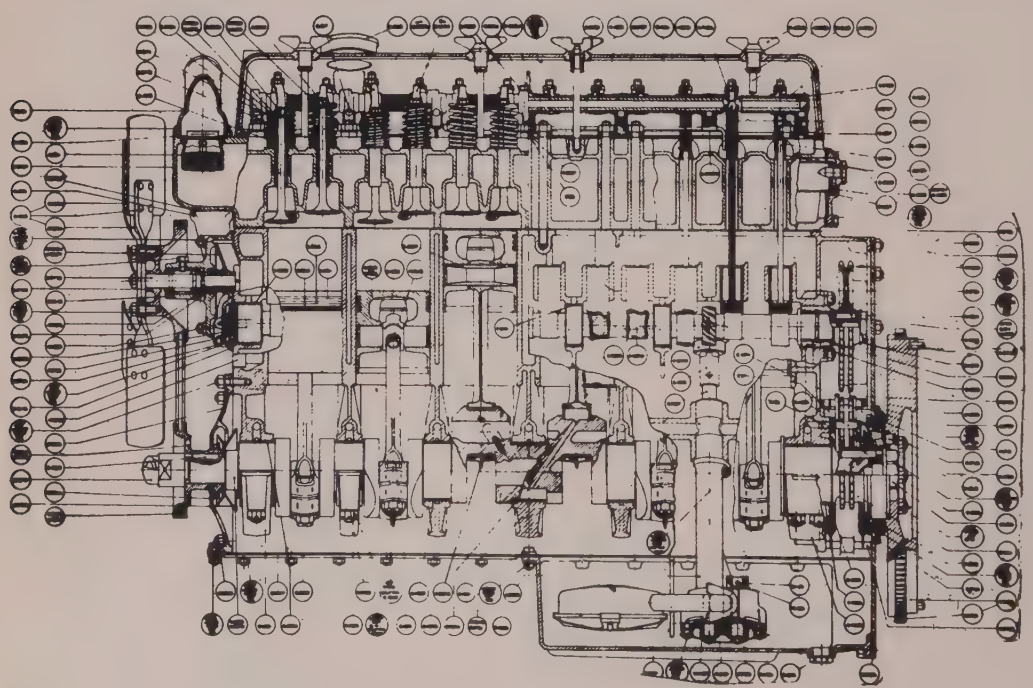

Fig. 1 – Alvis 6-cyl engine

from a torsional viewpoint, but with the nearly square cylinder proportions and fairly generous overlap between crankpins and main bearings achieved by using large diameter bearings, the disadvantages of length can be greatly overcome by increased section modulus.

The induction system is another weakness of the 6-cyl in-line engine because of the 120-deg spacing of the crankpins, which means that, if the valve events are over 240 deg, there is overlap between cylinders, so that two cylinders will be drawing charge from the induction system at the same time. It is almost impossible to design a cam form to give effective breathing with a 240-deg event to serve the engine up to 4500-5000 rpm which is the engine speed that is required these days for this type of car. Here, it must be said that the real usefulness of this engine speed could be debated.

By splitting the induction system into two, or three, the almost universally accepted firing order of 1-5-3-6-2-4 can be accommodated without one cylinder robbing another during its induction period. If intercylinder robbery of induced charge is prevented, equal bmep can be obtained for an in-line engine as against a single-cylinder unit of the engine.

The Alvis choice was originally a dual carburetor, each barrel having a separate induction pipe, one barrel feeding the front three cylinders and the other feeding the rear three cylinders. Later, a more orthodox arrangement was tried, consisting of two carburetors with separate induction manifolds having a balance pipe between them. The power output with these alternative arrangements was very similar and the engine is now fitted with two S. U. sliding choke H4 1½-in. horizontal carburetors.

The main structure of the engine – the cylinder block and crankcase unit – is cast iron, with a cast-iron cylinder head, carrying vertical in-line valves of orthodox arrangement. These valves are push-rod operated from a camshaft mounted rather high in the crankcase, mainly to achieve short, stiff knitting-needle-type push-rods operated by bucket-type tappets.

The cylinder barrels are all separated and surrounded by coolant.

The aluminum sump houses a submerged pump of the gear type with floating pickup incorporating a strainer unit.

The cylinder bore and stroke of 3.307 x 3.543 in. (84 x 90 mm) were, in some way, influenced by the modern trend toward square cylinder proportions, but, in order to retain some conservatism, the designers did not go immediately to a square engine, as this would have meant a slightly longer unit for the capacity.

Development has shown the advantages of keeping the stroke short and the designer would tend to reduce the stroke further on any future engine. For instance, this 6-cyl in-line engine appears to offer great advantages from a bearing point of view. The actual area of the crankshaft main bearings is practically half the total piston area. Bearing wear in practice has been shown to be very good, and there are no lubrication problems.

For this design it was considered that maximum efficiency and durability should be obtained in the range of speed within which the engine would spend most of its life, and that this was more important than odd bursts of a few minutes duration when maximum speed is permitted by better than usual road conditions.

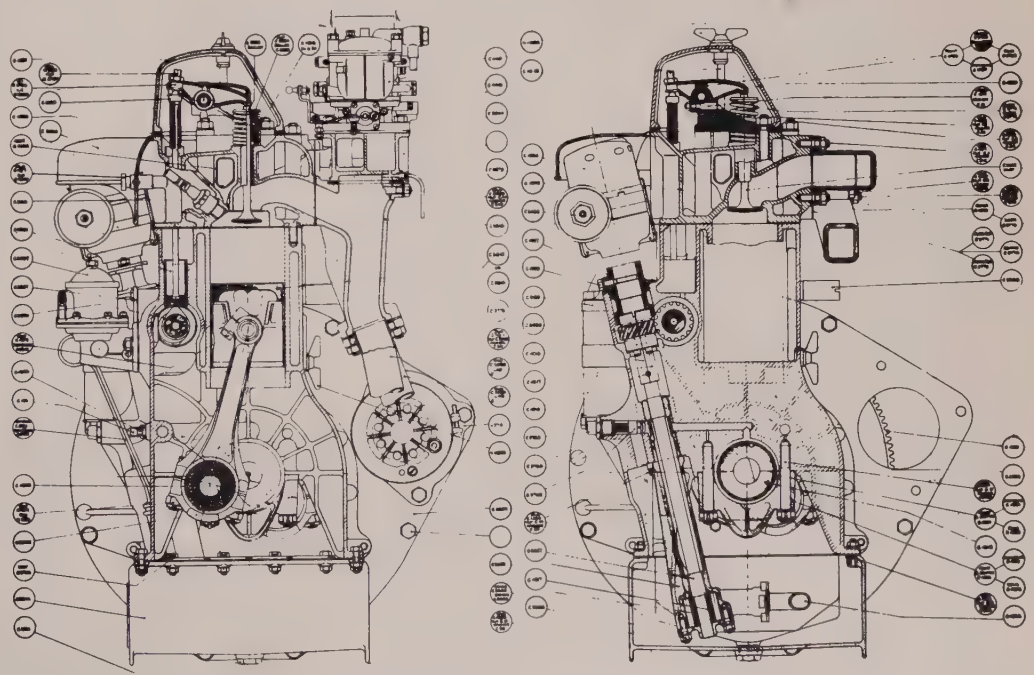

Fig. 2 – Alvis 6-cyl engine

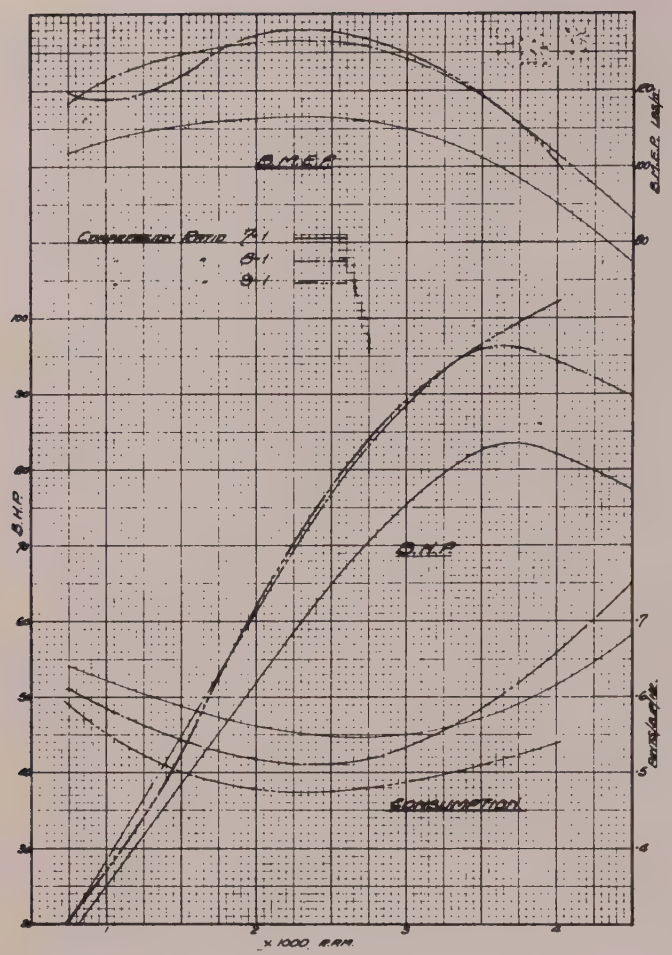

Fig. 3 – Performance curves for Alvis engine

From the above it was thought that the type of combustion chamber the Alvis Company had used for some 25 years, with slight modifications, was more suitable than those offering maximum output at high engine speed.

It is well known that the hemispherical head will give very much better power and better porting arrangements if the design is chosen for high-speed, high-power operation, but the Alvis designers think the design chosen shows up to advantage in the middle-speed range and at part throttle, in that it gives generally better combustion. The combustion chamber is simple, compact, and has some quench area. In considering other designs, the comparison, on a compression ratio basis, for general power and economy in the midspeed range seems to favor the compact shape with some quench characteristics. This design, which allows vertical in-line valves that are simple to install and operate, does not produce excessive turbulence or roughness and has good resistance to detonation.

The silchrome I intake valve has a throat diameter of 1.5 in. with a lift of 0.397 in. The austenitic exhaust valve has a throat diameter of 1.25 in. with 0.397-in. lift. Both valves have stems of 0.312 in. and run with 0.009-in. hot clearance, the valve timing being:

Inlet: Opens 30 deg early
Closes 68 deg late
Exhaust: Opens 71 deg early
Closes 25 deg late

The engine is fitted with aluminum-alloy pistons with two compression and one oil control ring, all above the piston pin. The piston-pin diameter is 0.9375 in.

The design of the crankshaft is considered a good compromise. Seven main precision-fit white-metal bearings were chosen because it costs no more to bore seven housings than one if multiple tools are used and, in a similar way, the milling of the bearing housings in the crankcase costs no more because the seven are milled in the same time that it would take to mill one. The advantage of the 7-bearing shaft is that there is great rigidity in torsion and in bending, which means smoother operation. There is no apparent increase in engine roughness between 7/1 and 9/1 compression ratio. The crankshaft is made of nodular cast iron. The proportions are such that no torsional vibration damper is required. The crankpins of 2-in. diameter and main journals of 2½-in. diameter are lightened by drilling so that the total weight of the shaft is 50 lb. No counterbalance weights are used, or have been found necessary, thus easing the torsional problems.

The drive to the camshaft is situated between the rear main bearing and the flywheel and is by means of a duplex roller chain. This, the designers consider, is the ideal position for the camshaft drive, as the nodal point of the crankshaft should be in the center of the rear main bearing. It is admitted that the firing order can vary this position, but it is thought best to have the drive to the camshaft as near the nodal point as possible to obviate the transfer of torsional oscillations to the camshaft. It can be argued that the rear location of the timing gears is much more inaccessible than the more usual front-end location, but there can surely be no objection to anything one can fit and forget. It is usual that these rear-mounted duplex roller timing chains will last 250,000 miles without attention.

A spring-loaded idler pulley is used as a tensioner, owing to the camshaft being placed as high as possible in the cylinder block, necessitating a rather long chain. A spring-loaded tensioner is necessary to take care of the growth in general chain length, due to wear. The real trouble from the chain is the number of free links between the wheels dancing to their own harmonics, which are determined by the number of these links. It was found necessary to put a damping pad on the free side of the chain to prevent the full amplitude of chain vibration causing noise.

The crankcase gains rigidity from having seven main bearings and, therefore, seven walls supporting these bearings. As these are corrugated or ribbed they add to the general rigidity of the structure. Thus, wall thickness of the structure can be kept as thin as practical casting permits, and it is

unnecessary to resort to heavy horizontal flanges to absorb transverse vibration.

The lubrication of the main bearing housings is by means of a rectangular groove machined in the center of the housing face. The bearing shell, when fitted into the housing, forms a seal on top of this groove. Oil enters the shell through four holes spaced around the circumference of the shell. This is thought to be the correct method of feeding lubricant to the crank journal, as the journal tends to move around the bearing shell eccentrically, and therefore the point of lowest pressure is varying as it rotates. Thus, the oil can be continually fed from the point of lowest pressure, which means a continuous supply and more oil actually passing through the bearings with this circular groove feed. The pump capacity needs to be increased to maintain the flow with this system as against the more orthodox direct feed.

The crankpins, in distinction to the main journal bearings, do not have a rotating maximum load center, the maximum load always occurring on the inside of the crankpin. Therefore, the view of the designer is that the correct oil supply entrance is toward the outside of the pin, the actual point of oil entry to the big end bearing being 30 deg in advance of the maximum radius point.

To justify the lubrication system, this engine has very good resistance to wear of its bearings, and engines have been raced in sports cars running on SAE 5 lubricants with no sign of bearing distress.

No filter is used in the lubrication system other than the screen in the floating type of strainer in the sump. The designers think that, provided built-in dirt can be filtered out of the engine during its initial running-in process, it is unnecessary to filter the oil during running, and argue that about 99% of the residue collected in the filters is entirely harmless to the bearings.

The engine is pressure water cooled by a vane-type pump having the radiator fan at the opposite end of its spindle and being driven by a V belt from a pulley mounted on the nose of the crankshaft.

The exhaust system is split into two 3-cyl units.

With its standard 7/1 compression ratio, the engine gives 117 bmep between 2000 and 2750 rpm; 52 hp at 2000 rpm; and a maximum of 84 hp at 3750 rpm. With 9/1 compression ratio the engine gives 62 hp at 2000 rpm; 136 bmep between 2000 and 2500 rpm; with a maximum power of 102 bhp at 4000 rpm. (See Fig. 3.)

The whole aim of the design has been to produce a rugged, reliable engine which will give long life with little attention and will have good power and throttle response in the car in the middle-speed range, combined with good fuel usage. No attempt has been made to squeeze the design for maximum output.

Armstrong Siddeley Sapphire Engine

The Armstrong Siddeley 3.4-liter, 6-cyl, in-line Sapphire engine manufactured by Armstrong Siddeley Motors Ltd., of Coventry, has a bore and stroke of 90 mm (3.543 in.) giving a cubic capacity of 209.636 cu in. (See Figs. 4 and 5.)

The standard compression ratio is 7/1, the combustion chamber being hemispherical, and the push-rod-operated overhead valves being inclined at 70 deg (35 deg each side of the vertical centerline).

The choice of an in-line arrangement for the cylinders was dictated mainly by the policy of keeping the hood line narrow and the overall width of the

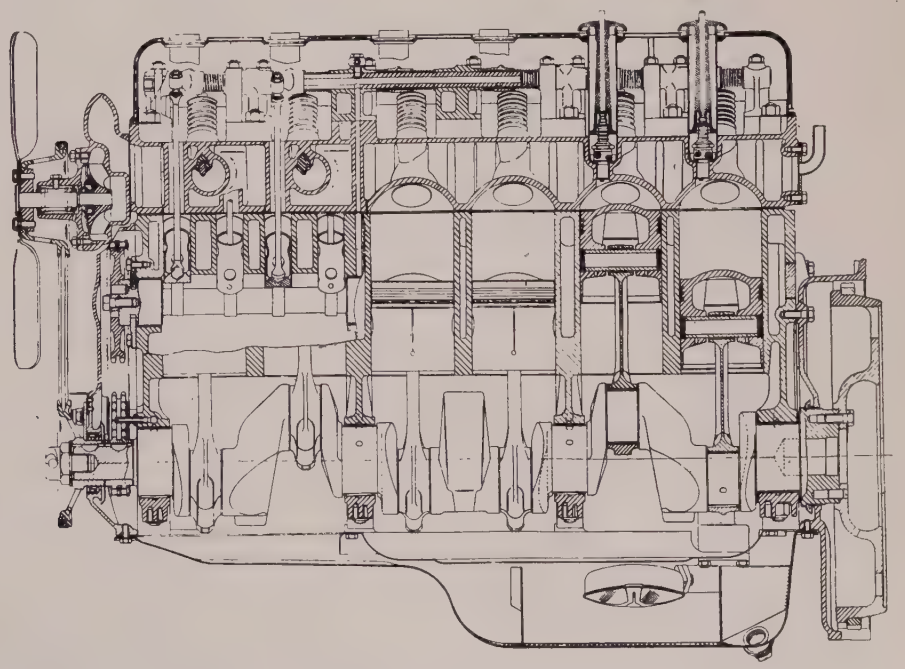

Fig. 4 – Armstrong Siddeley Sapphire engine

car as sleek as possible. A secondary consideration was that the machine tool equipment, most of which had been replaced since the war, was more suitable for the manufacture of 6-cyl in-line units.

The hemispherical head was chosen after exhaustive tests of this arrangement and the bathtub type, which was tested with and without squish. The tests showed that the breathing arrangement of the hemispherical type was so superior that the question settled itself. An increase in power of 12½% was obtained for the same compression ratio and engine speed with a lower octane requirement, mainly due to the better cooling of the combustion chamber, and the combustion smoothness appeared to be at least as good as with the squish head.

The cylinder head is an iron casting. The silchrome I inlet valves are of 1.55-in. throat diameter, 0.34-in. stem diameter, and have a 0.358-in. maximum lift. The exhaust valves are made from XB steel and seat directly on to the iron head, having a 1.35-in. throat diameter, 0.34-in. stem diameter, and 0.3075-in. maximum lift. The standard valve timing is:

Inlet: Opens 8 deg early
Closes 62 deg late
Exhaust: Opens 46 deg early
Closes 18 deg late

The valve timing was obtained after numerous tests to get the best compromise between maximum power and a good, low-speed torque. The effect on fuel economy was also carefully considered.

It will be seen from Figs. 4 and 5 that the inlet manifold consists of a long, machined passage, which is almost exactly half buried in the head, into which run the six inlet ports, which are all of equal length.

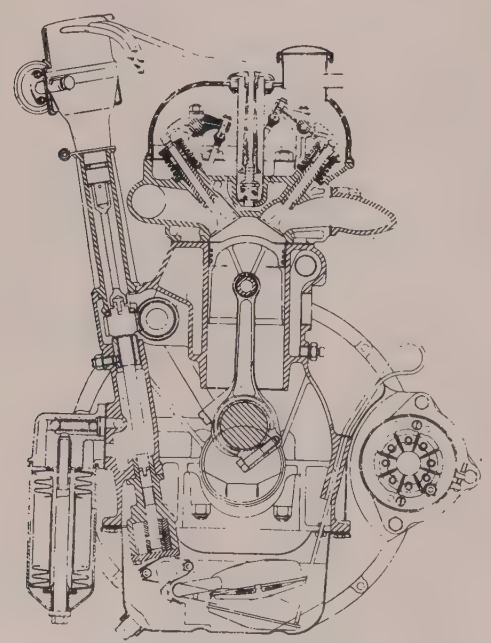

Fig. 5 – Armstrong Siddeley Sapphire engine

The outer wall of the manifold has four equally spaced flanges, to which is nominally connected a straight rake water heated manifold for the single Stromberg 36-mm carburetor or the two separate water-heated manifolds for the twin-carburetor arrangement. This arrangement has proved very useful, as it has enabled a number of carburetor and induction manifold changes to be carried out in order to obtain various engine ratings, which, in the case of the standard cars, are 120 bhp for the single-carburetor installation and 150 bhp for the standard 2-carburetor installation. Considering possible future development of this unit, over 200 bhp has been obtained with two special carburetors and a power output considerably in excess of this figure with four special carburetors and other modifications.

It will be seen that both inlet and exhaust valves are operated from a single camshaft placed fairly high in the cylinder block. To compensate for the weight of the unequal-length push-rods, the longer rod is fitted to the smaller exhaust valve. The camshaft is driven by a duplex chain fitted with a tensioner of the idler pulley type mounted on an eccentric spindle, as experience has shown that this type is more reliable and less noisy than the automatic spring-loaded adjuster.

The cylinder block and top half of the crankcase is of unit construction in cast iron without cylinder liners, the oil pan being a mild-steel pressing.

When deciding on the bore/stroke ratio of unity, the main idea of the designer was to keep the piston speed low. This works out to be 2480 fpm at approximately 85 mph with the standard axle ratio. It was also thought that some slight sales value could be obtained by pandering to the present fashion for square engines. The results appeared to have been very satisfactory, the bore wear figures taken from a number of experimental cars showing a maximum of 0.0018 at 76,000 miles.

The crankshaft is a forging made from 55-ton (250 Brinell) steel, with four main bearings of 2.75 in. diameter, the crankpin diameter being 2.125 in. White-metal strip bearings of the precision-fit type are used for the journals and connecting rods, the balance weights being integral. Oil is fed to the big ends by a plain drilled hole from the main journal.

The connecting rods are 40-ton (180 Brinell) steel forgings, the rod centers being 6.5 in., the L/S ratio 1.825, and the bolt diameter is 0.375 in. The piston pin is 0.875-in. external diameter, bored out to an internal diameter of 0.625 in.

No crankshaft damper is fitted. The maximum amplitude recorded in the running range is:

±0.35 deg for 6th order at 3500 rpm
±0.25 deg for 4½ order at 4680 rpm

The natural frequency is 21,000 cpm.

Fig. 6 shows the performance of the engine with various induction and camshaft arrangements. The broken lines represent the engine in standard tune

(manufacturer's setting) with one carburetor. The next solid line shows the same engine with twin carburetors. The top solid line shows the peak performance achieved with 8/1 compression ratio, special camshaft, and twin carburetors.

Austin A.40 Engine

The Austin A.40 engine (Fig. 7) was developed in the immediate postwar era as a light passenger-car engine by the Austin Motor Co. Ltd., of Longbridge, Birmingham.

The makers think that this arrangement of a simple, vertical, water-cooled 4-cyl in-line engine is the best solution for a small car of good quality (gross weight of under one ton – 2240 lb).

In general, the engine is an orthodox valve-in-head unit of 2.578-in. bore, 3.5-in. stroke, giving a capacity of 73.8 cu in.

This proportioning gives a short engine with compact combustion chambers, with coolant surrounding each cylinder bore. With 3.5-in. stroke, it cannot be considered a long-stroke engine, since at 5000 rpm the mean piston speed is under 3000 fpm.

The compact engine achieved permits a short, stiff 3-bearing crankshaft, camshaft, and rocker shaft, all of which influences the satisfactory operation of the valve gear. The cylinder head and unit-construction cylinder block are in cast iron, the sump being a steel pressing. The layout of the engine is designed to provide a simple and rigid structure, which will be easy to manufacture and maintain. Action has been taken in the design, wherever possible, to avoid complications such as tubes pressed into the cylinder head to provide for the push-rods and large cover plates on the block.

The engine is fitted with one Zenith 30 VIG-8 down-draft carburetor with accelerator pump and economy device, and with manual choke operated from the dashboard. The carburetor receives its fuel from a camshaft driven diaphragm-type pump.

The induction pipe is an aluminum casting bolted to a hot spot at the center of the exhaust manifold. A stainless-steel hot-spot plate separates the induction and exhaust systems. This plate has a heat pickup extension into the main exhaust gas stream to increase the heat intensity at the center of the hot-spot plate, which is directly under the main gas stream from the down-draft carburetor. This arrangement provides a simple static induction heat system giving quick warmup without excessive running heat. The arrangement is foolproof in that it cannot stick one way or the other, as can happen with mechanical thermostatic hot spots. The induction pipe is kept as short as possible with straight arms feeding into the siamesed inlet ports.

The inlet ports are arranged to give a directional swirl turbulence through the intake valves into the heart-shaped combustion chambers, which lie on either side of their respective siamesed induction ports.

Part of the combustion chamber is formed in the concave crown of the piston to promote swirl turbulence further. This form of combustion chamber is a Weslake patent. It is claimed it gives (a) high turbulence, (b) good resistance to detonation, (c) good filling and scavenging, and (d) simple construction. The peak provided in the combustion chamber by the apex of the heart reduces heat radiation from the exhaust valve to the intake valves. This helps cylinder filling and some squish action is provided below this peak in the combustion chamber.

The cylinder-head layout is unorthodox in that the push-rods for operating the valves pass up the same side as the siamesed ports. This allows the spark plugs to be screwed into the opposite side of the combustion chamber from the push-rods, and thus the environment of the spark plug is free from restriction and a good job of cooling the plug with unrestricted water flow can be done. Further, the

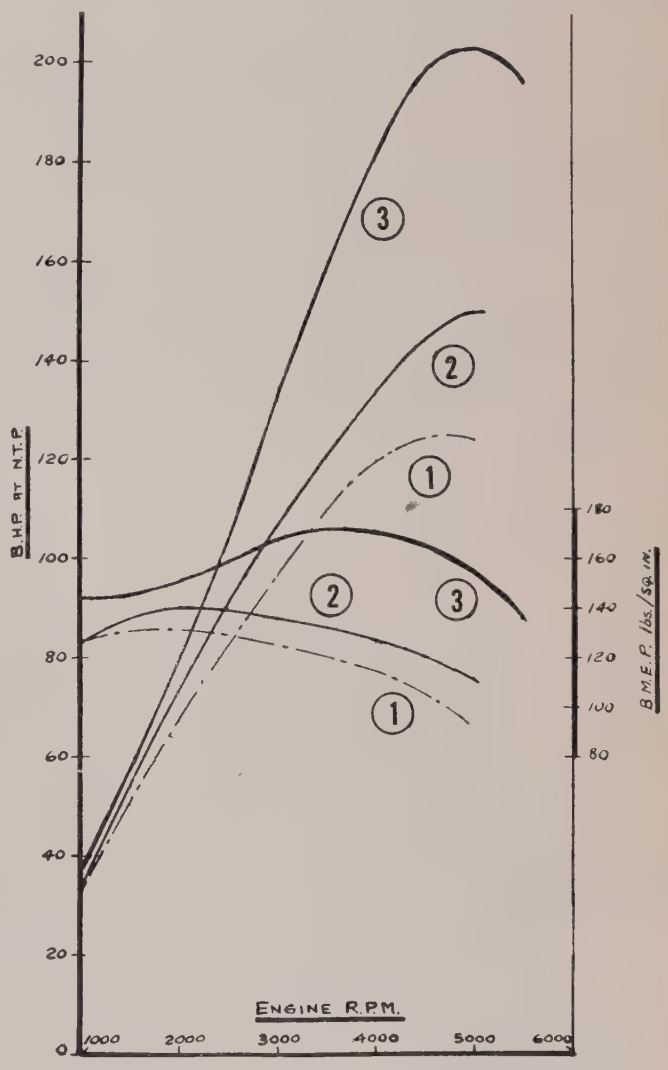

Fig. 6 – Performance curves for 3.4-liter Sapphire engine
(1) Standard engine with one carburetor
(2) Standard engine with twin carburetors
(3) Engine with 8/1 compression ratio, special camshaft and twin carburetors

plugs are located in the cylinder head at a very accessible angle, and servicing the plugs is a simple job.

Apart from the above, the engine is of orthodox layout, although it incorporates a number of details of Austin practice. The chief of these are:

1. Lubrication of the crankshaft white-metal precision-fit main bearings is by machined annular grooves in the crankcase and bearing cap, which provide radial feed holes into the main bearings.

2. The big end bearings fitted in forged steel connecting rods have a patented lubrication system, an extra drilled hole being provided on the trailing side and toward the inside of the crankpin so that the wipe of the connecting-rod bearing distributes lubricant over the inertia load side of the pin. An additional hole with controlled flow by fitted jet is provided in the outside of the crankpin at the maximum radius point. This arrangement has been found to give good bearing life and very even wear over all the crankpins.

3. Apart from the connecting-rod-bearing lubrication system, no calibrated jet holes are provided for lubricating other parts of the engine, low-pressure control being achieved by holes matching at intervals on rotation; thus, these holes are unlikely to become choked. This scheme is applied to the lubrication of the complete rocker-shaft assembly and to the timing chain sprocket and camshaft thrust plate. Oil pressure from the hollow rocker-shaft is applied to the rocker bearings and to the tips of the spherical push-rod ball pins, overflow from which lubricates the lower end of the push-rods and tappet tops. A bleed hole is provided to give controlled oil feed to the finger end of the valve rockers.

4. A separate passage gives pressure lubrication to the tappet barrels.

5. As the camshaft is driven by a duplex roller chain, some means of chain tensioner has to be provided. This is done by a rectangular-section synthetic rubber tensioner ring, which is placed on the camshaft sprocket in a machined groove between the two rows of teeth. This provides a tensioner that rotates with the chain.

6. Oil is prevented from passing down the valve guides by synthetic rubber sealing grommets, which are fixed at the valve spring end of the guide.

The intake valve is of 1 3/16-in. throat diameter, with a 5/16-in. stem diameter. It is made in silchrome I steel with a 45-deg seat. The exhaust valve is of 1-in. throat diameter with a 5/16-in. stem diameter. It is made of XB steel with a 45-deg seat. The valves run with 0.015-in. tappet clearance, and there seems to be little change in tappet clearance through the engine operating temperature range. Both valves have 5/16-in. maximum lift and are fitted directly into the cylinder head without seat rings. The exhaust-valve seats themselves are 0.100-in. wide. It has been found that these rather wider than usual valve seats give good life and reduce the tendency of the engine to after-run.

The valve timing is:

Inlet:	Opens 5 deg early
	Closes 45 deg late
Exhaust:	Opens 40 deg early
	Closes 10 deg late

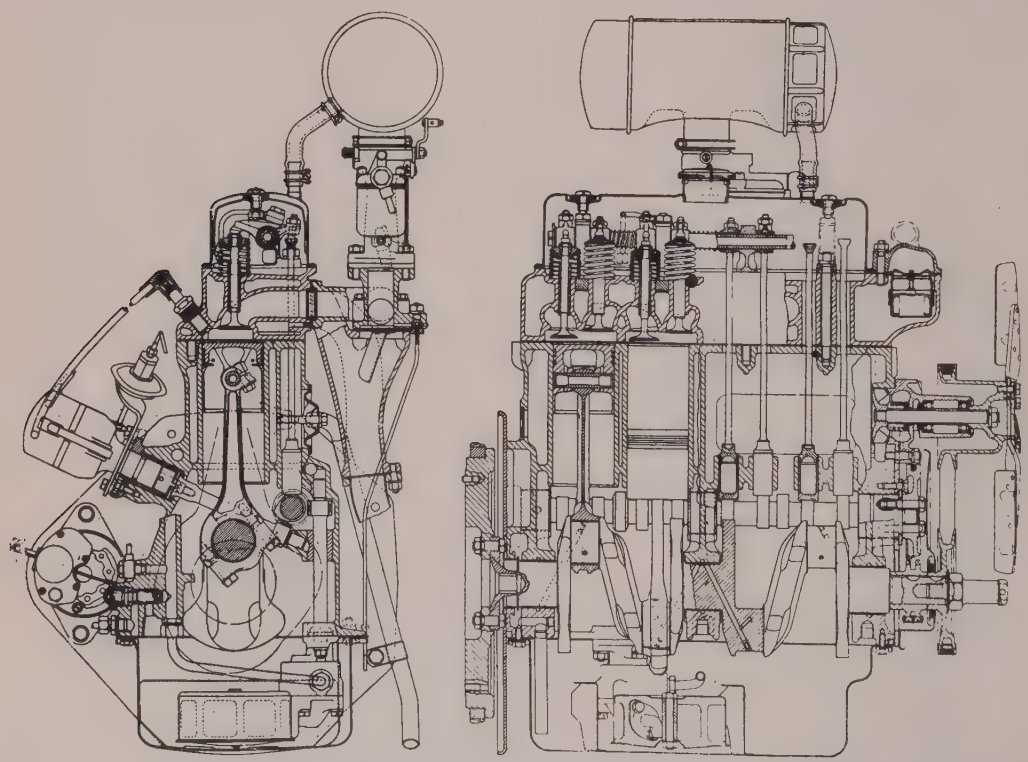

Fig. 7 – Austin 4-cyl overhead-valve engine

The above timing is with 0.021-in. clearance at the valve. After timing, clearance is reset to 0.105 in.

The exhaust-valve guides are counterbored at the valve head end to prevent valve stick. It has been found in practice that the valve gear is not sensitive to highly leaded fuels.

The cam form (of Austin patent design) avoids any rapid changes in acceleration and deceleration. The pistons are made in LO-Ex (low-expansion) aluminum alloy anodized. Three compression rings and one slotted scraper ring are fitted above the piston pin, which is clamped into the little end of the connecting rod. The second compression ring is tapered for quick bedding and effective oil control early in the life of the engine.

The cylinder bores are individually fine bored to size to give a granulated surface, the crests being removed by a wire brushing process. Oil retaining pockets are thus provided throughout the length of the cylinder bores, which helps the running-in process.

The 3-bearing forged-steel crankshaft is carried on precision-fit white-metal bearings. It is of sturdy proportions with over 50% integral counterbalance to relieve the load on the center main bearing. A submerged gear type of oil pump driven from the camshaft is situated in the sump. The oil intake to the pump is through a large fixed strainer, and oil is delivered to the engine through a very adequate bypass oil filter, which is integrally mounted with its bracket on the right-hand side of the engine in the vertical position. From the oil gallery, oil is delivered to the main, big end, and camshaft bearings under a running pressure of 50 psi. The connecting rods have small bleed holes to provide oil for the cylinder walls when starting up. The front camshaft bearing feeds oil to the camshaft gear for timing chain lubrication. The rear camshaft bearing feeds oil to the overhead rocker gear and provides the pressure feed to each tappet. In general, oil returns to the sump from the valve gear, via the push-rod apertures. There are no external oil pipes on the engine. The engine carries seven pints (8.4 U. S. pints) of oil in the sump.

The cooling system is quite orthodox, the pump being driven by a V belt from the front end of the crankshaft and being fitted with a packless-type gland.

Performance curves for the engine are given in Fig. 8. At 2400 rpm the engine produces 59 ft-lb torque; 120 bmep; at 4500 rpm it produces 42 bhp, 102 bmep. These figures are corrected for standard barometric pressure and 60 F air intake with test house exhaust system.

Humber Blue Riband Engine

At the termination of World War II, the Rootes Group, makers of Humber, Hillman, Sunbeam-Talbot passenger cars and Commer commercial vehicles, formulated an engine policy to cover their postwar program.

High on the list was the development of a basic 6-cyl engine that would not only serve the needs of a wide range of commercial vehicles, but could be extended to cover the more refined requirements of a high-quality passenger car of between 3800 and 4450 lb gross weight.

This unit (Figs. 9 and 10) can be said to be an orthodox 6-cyl in-line, push-rod, valve-in-head layout. The main structural members of the engine, the cylinder head, and cylinder block crankcase unit are cast iron, the valves are arranged vertically in the cylinder head and in line down the engine, and the crankshaft is carried on seven main bearings.

As the commercial vehicle program alone ranged from 3- to 12-ton models (long tons), including fully forward control and engine in front of driver types, the demands appeared rather exacting. The terms of reference for the engine were set out as follows:

1. It must lend itself to easy installation in a wide variety of vehicles.
2. It must be strictly a basic engine in that the majority of its components should be interchangeable from one application to another.
3. Wherever noninterchangeable parts were unavoidable because of the particular characteristics of a vehicle, these must be designed to fit to the basic unit with a minimum of complication.
4. Noninterchangeable parts must, wherever possible, be capable of being machined and assembled on existing jigs, tools, and fixtures.
5. Whatever the application, the specification must produce a highly competitive performance within the requisite classification.

It was realized, in formulating the above, that a number of years would elapse before the full policy could be implemented and the maximum benefit derived.

In 1947, the first step toward translation of the above policy into practice was taken, with the introduction of the Commer 5- to 7-ton fully forward control range of commercial vehicles, powered by a

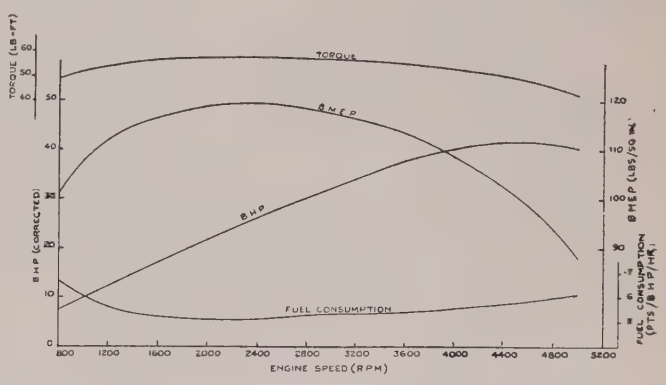

Fig. 8 – Performance curves for Austin engine – 7.2 compression ratio

semihorizontal 3.75-in. bore version of this engine.

Equipped with chromed cylinder bores, this first application of the Rootes Group 6-cyl engines fulfilled the hopes and objectives that had been planned for it.

The second stage of the program, that of producing a vertical version of the semihorizontal engine, was embarked upon against an already established background. Development was quickly completed and a 3.75-in. bore vertical engine became available for use in the more conventional forward engine range of commercial vehicles. This now provides the power unit for the present Commer 3-ton, 4-wheel drive, which is in production for both civil and military use. Subsequent development catered to a commercial range to cover lighter vehicles and in 1951 a 3.5-in. bore vertical engine was introduced for such vehicles.

In 1952 a refined version of the 3.5-in. bore vertical engine, called the Blue Riband, was produced to meet the requirements of the latest Humber Super Snipe and Pullman passenger cars.

This is a very brief history of the four main variations of the Rootes Group 6-cyl engine.

The passenger-car version of this 6-cyl engine has a bore of 3.5 in. and a stroke common to all versions of the engine of 4.375 in., which gives a displacement of 252.6 cu in.

The cylinder block and crankcase form a compact one-piece iron casting, embodying full-length water jackets, the whole structure being stiffened by the five crankcase webs, which, together with the front and rear end walls, support the seven crankshaft bearings. The crankcase walls extend 3½ in. below the centerline of the crankshaft. Each cylinder bore is completely surrounded by coolant and has a wall thickness of 0.25 in. Lateral location of the cast-iron bearing caps is effected by recessing them 0.3125 in. into the webs and end walls of the crankcase. These bearing caps are held in position by 0.50-in. steel studs and self-locking nuts. The stud positions are asymmetrical so that the bearing caps cannot be assembled wrongly.

The cylinder head has fully machined offset bathtub combustion chambers and all separate valve ports. The casting is open ended so that the cores may be easily removed after casting, the end holes being covered with bolt-on pressed-steel cover plates. Studs of 14 x 7/16-in. diameter secure the head to the cylinder block, the joint being made with a copper and asbestos gasket.

The valves are installed directly into the cylinder head, without seat rings, the intake valve being made of silchrome I steel having a throat diameter of 1.60 in., a stem diameter of 0.373 in., and a lift of 0.41 in. The valve stem is shrouded and sealed against oil loss at the valve spring end of the valve guide. The exhaust valves are made of XB steel, have a throat diameter of 1.378 in., a stem diameter of 0.373 in., and a lift of 0.41 in. Valve timing is:

Inlet: Opens 15 deg early
Closes 57 deg late
Exhaust: Opens 57 deg early
Closes 15 deg late

The valves are operated from a side camshaft, through barrel-type tappets, tubular push-rods, and rocker arms of 1.3/1 ratio. Adjustment is provided by hardened ball-headed screws in the valve rockers.

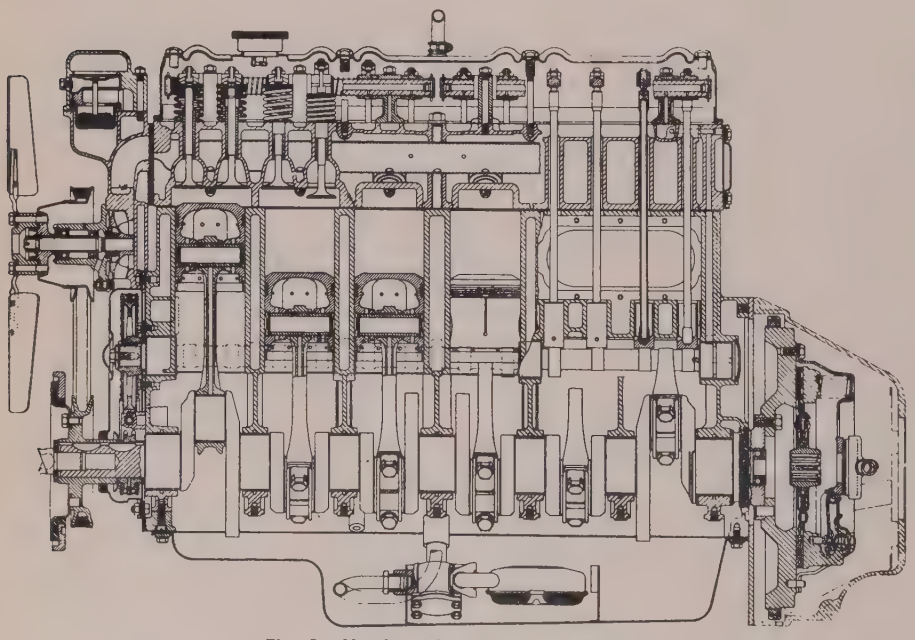

Fig. 9 – Humber Blue Riband 6-cyl engine

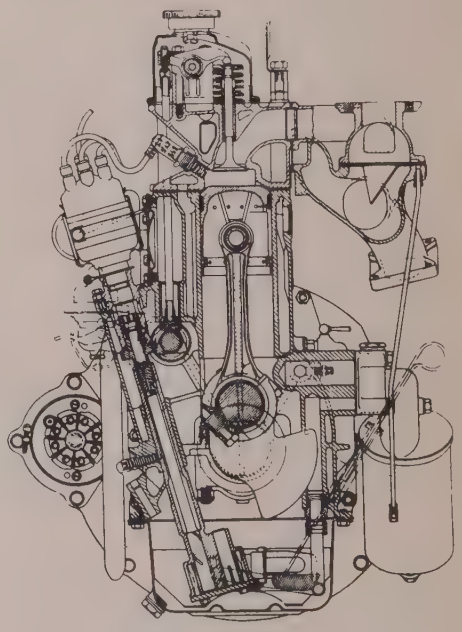

Fig. 10 – Humber Blue Riband 6-cyl engine

The camshaft runs in four renewable steel-backed microbabbitt precision-fit bearings, being driven by a duplex roller chain from the front end of the crankshaft, the chain being tensioned automatically by a hardened steel chain tensioner of the blade and slipper type, carried on a spring-loaded plunger. This tensioner bears on the inner face of the chain and is assisted by centrifugal force in maintaining the correct degree of chain tension when the engine is running. The camshaft incorporates a skew gear for driving the oil pump and the distributor, and an eccentric opposite No. 1 cylinder to operate the diaphragm-type fuel pump.

The forged crankshaft in 55-ton (250 Brinell) steel, is counterbalanced by integral counterweights to relieve the load on the main bearings, which are steel-backed precision-fit white-metal-lined bearings of 2¾-in. diameter, the crankpins being of 2¼-in. diameter. Torsional rigidity of the shaft is assisted by these dimensions, which result in an overlap between the main and crankpin journals.

The H-section connecting rods produced from steel forgings have lead-bronze bushings to carry the 1⅛-in. diameter hollow piston pins, which are full floating and secured by circlips (piston-pin spring-retaining clips) in the aluminum pistons. The connecting rods are rifle drilled to feed oil under pressure from the big end bearings to the piston pins. The big end bearings are split diagonally at 45 deg so that the connecting rods may be withdrawn through the cylinder bores for servicing, the bearing caps being secured by two 7/16-in. bolts locked by tab washers (a type of lock washer), the rod centers being 8¼ in., which gives a rod length to stroke ratio of 1.885.

The pistons are of the die-cast T-slotted type in Lo-Ex (low-expansion) aluminum alloy, each having two compression or gas rings, and one oil control ring, above the piston pin, and provision for a second oil control ring at the bottom of the skirt below the piston pin. For this passenger-car engine the top ring is chromium plated, the face width of the ring being 0.0928 in. for the compression rings and 0.1875 in. for the oil control ring, the radial thickness of the rings being 0.138 in.

The lubrication of the engine is by a large gear-type oil pump driven from the camshaft submerged in the engine sump supplying oil to the crankshaft, camshaft, and valve gear, through drilled holes from the main oil passage. There are small holes in the top half of the big end bearings to direct oil jets on to the cylinder walls. Oil conditioning is effected by a strainer in the sump at the pump intake, which is of the floating type, and a large full-flow filter in the pressure pipeline with a pressure relief valve in the filter housing.

The mixture from the carburetor is distributed to the six separate ports in the cylinder head through a 6-branch aluminum induction manifold with a stainless-steel bolt-up type of hot-spot plate, which forms the junction between the intake manifold and the exhaust manifold at the center of each, and directly under the carburetor.

Coolant is circulated directly from the centrifugal pump mounted at the front end of the engine into the cylinder head, and is distributed through a distributor tube, which goes the full length of the cylinder head and is ported so that the coolant is directed, in the form of jets, on to the exhaust-valve ports and spark-plug bosses, circulation around cylinder being by thermosiphonic action.

Ignition is by 12-v coil and distributor, the latter incorporating builtin suppression equipment and fully automatic control. 14-mm long-reach spark plugs are used with 0.030 in. gap. The generator is mounted at the front end of the engine and is driven by a V-belt triangulated drive, which includes the water pump drive in the orthodox manner. The driving pulley, mounted on the front end of the crankshaft, is formed integrally with a rubber-sandwiched torsional vibration damper, which deals with high-speed torsional vibration of the crankshaft.

With 7.13/1 compression ratio, this engine develops (Fig. 11) 116 hp at 3600 rpm, and 211 ft-lb of torque at 1400 rpm. The maximum bmep is 126 psi at 1400 rpm. A minimum brake specific fuel consumption of 0.55 pints/bhp/hr (one British pint = 1.2 U. S. pints) is obtained at 2000 rpm. The dry weight of the engine is 758 lb. The overall

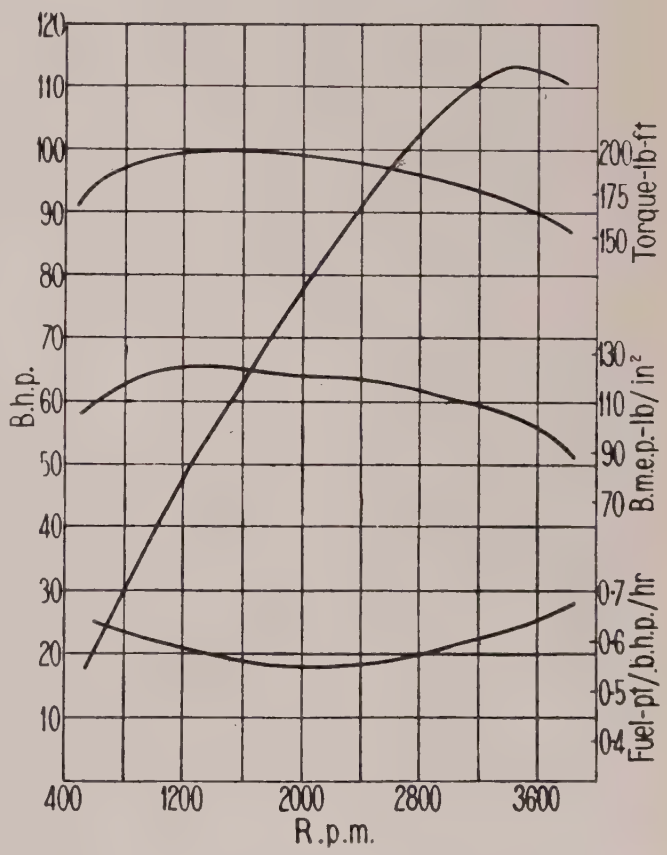

Fig. 11 – Performance curves for Humber Blue Riband engine

dimensions of the engine are 38 in. long x 29½ in. high x 22 in. wide.

Jowett Engine

The Jowett horizontally opposed 4-cyl liquid-cooled engine (Figs. 12 and 13) is produced by Jowett Cars Ltd., of Idle, Bradford.

The design of the engine was originated in 1945 for the Jowett Javelin 4-seater sedan passenger-car of one-ton gross weight, which was being designed from a clean sheet of paper at that time.

To obtain good riding comfort, the designers wished to seat the passengers as near the center of the car as possible; this resulted in having the feet of the front passengers resting between the front wheel arches, and thus, very little room was left for the power unit in view of the chosen car wheelbase of 102 in. The obvious choice of a power unit layout seemed to be a flat engine with opposed cylinders with the radiator mounted behind it.

Having fixed the basic car layout, the designers decided to develop a flat 4-cyl unit of 90.9 cu in. capacity to develop 50 hp to give the car a maximum speed in excess of 80 mph.

The cylinders are horizontally opposed in pairs, with a bore and stroke of 2.85 x 3.54 in., giving a stroke/bore ratio of 1.24/1. The firing order is 1-4-2-3, the cylinders being numbered from the front in the order in which they are connected to the crankshaft, that is, the left-hand pair being Nos. 1 and 3, and the right-hand pair Nos. 2 and 4. The total engine length achieved is 17.5 in. without transmission, and with the bulk of the transmission, 30 in. The engine width is 32.5 in., and total height to top of thermostat housing 26 in., the crankcase being 13 in. deep.

The crankcase is formed in two aluminum-alloy die castings split vertically on the crankshaft axis, each half carrying its integral cylinder block, to which wet centrifugally cast cast-iron liners are fitted. The coring of these die castings is comparatively simple, as the cylinders themselves are removable and the crankcase is open at the bottom to receive the pressed steel sump.

The cylinder liners are made of high-duty manganese chromium iron with a surface hardness of 270 to 280 Brinell, being located by flats on the bottom flanges. The top half of the liner is exposed directly to the coolant, the bottom half being spigot mounted into the crankcase casting (see Fig. 13), which spigot, in itself, is partially jacketed by the coolant. The bottom seal for the liners is formed by a flange approximately half-way down the liner seating on a packing ring on the spigot face. The top seal is formed by the cast-iron cylinder head being pulled down on a copper and asbestos gasket, which seats on the top face of the liner; thus, the top half of the liner is in compression and the bottom half is free to slide in the spigot cylinder.

The crankcase halves are held together by six main bolts situated either side of the three crankshaft main bearings, and five smaller bolts along the top flange, the center and rear bottom bolts being throughbolts. The design configuration does not allow the other main bearing bolts to be throughbolts. Accurate location of the two halves of the crankcase is achieved by two dowels above the front and rear main bearing faces.

The 3-bearing forged-steel crankshaft is machined all over with integral counterbalance weights, being carried on precision-fit trimetal bearings. The rear bearing is flanged to take the end thrust. The main bearings are 2.25 in. in diameter, the crankpins 2 in. in diameter and 1 in. wide. There is substantial overlap between the mains and crankpins, and both mains and crankpins are induction surface hardened to around 500 Brinell, finished to 8-12 micro-in. The width of the bearings is somewhat dictated by the cylinder centers, which

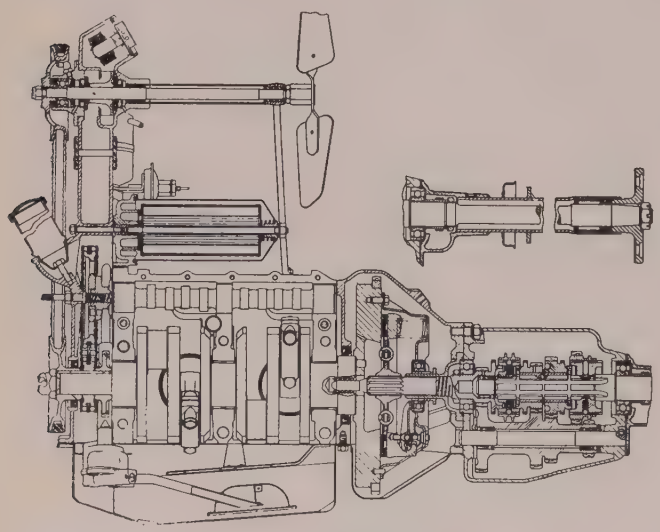

Fig. 12 – Jowett Javelin engine

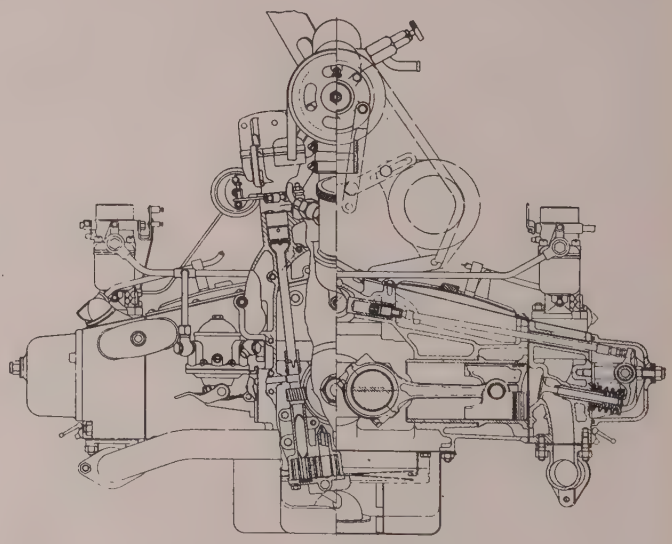

Fig. 13 – Jowett Javelin engine

are set at 5 1/16 in., with an offset between the opposing sides of 1 11/16 in. The cast-iron flywheel is registered on the rear end of the crankshaft located by a single dowel and secured by four setscrews, the steel starter ring being shrunk on.

The connecting rods are H-section steel forgings and are 6¼ in. between centers. To allow the rod and piston assembly to pass up the cylinder bore, the big ends are split at an angle of 45 deg with serrated faces for accurate location. The caps are secured by two 0.375-in. setscrews, tightened to 400 in.-lb torque, and locked with tab washers. The big end bearings are made with a similar technique to the crankshaft main bearings. The small ends are bronze bushed for the fully floating hollow-bored piston pins of 13/16-in. diameter, which are located in the aluminum-alloy pistons by circlips.

The pistons are split-skirt low-expansion aluminum alloy. They carry three *D*/26 rings, the top two of which are compression rings, all being above the piston pin. The top ring is chromium plate, taper faced and the second ring internally stepped to assist oil control by the bottom oil control ring. The pistons are camground and tin plated, the crown of the piston being dished to form part of the combustion chamber.

The cylinder heads are cast iron, each head being retained by eight 7/16-in. studs. The layout of the combustion chamber is a semipancake arrangement, being offset toward the top of the engine with the valves set-in in-line at 8 deg to the horizontal. As each cylinder head deals with two cylinders, the center valves are the intake valves, being fed from a siamesed induction port, which connects directly to one of the down-draft carburetors, a pair of carburetors being mounted one above each cylinder head.

The exhaust ports are arranged to exhaust vertically downward into short cast-iron manifolds, which are connected through a crossover arrangement to the main exhaust system, which is on the right-hand side of the car.

The design of the cylinder head is such that the individual exhaust valves are adequately cooled. As there is virtually no induction pipe for either carburetor, the siamesed induction port apex functions as the hot spot.

The intake valves are made of silchrome I steel, having a throat diameter of 1¼ in. and a stem diameter of 5/16 in. The exhaust valves are in XB steel, with a throat diameter of 1 3/32 in., with a stem diameter of 5/16 in. A seat angle of 30 deg is used for the intake valve and 45 deg for the exhaust valve. Valve lift is 0.315 in. for both valves. Valve timing is:

Intake: Opens 12 deg early
Closes 53 deg late
Exhaust: Opens 50 deg early
Closes 15 deg late

14 mm short-reach spark plugs are mounted on the top side of the cylinder head above the exhaust valves, being fully weather protected by rubber connectors, which seal on the insulators.

The valves are operated from a high-duty cast-iron camshaft located above the crankshaft and driven by a duplex roller chain from the front end of the crankshaft, without a chain tensioner. The chilled cams have a minimum surface hardness of 45 Rockwell C and operate onto flat-faced chilled iron tappets of similar surface hardness with a surface finish of between 15 and 20 micro-in. Early engines were fitted with automatic zerolash tappets, but these have since been replaced by fixed tappets, because, in general, the British motor industry was not sufficiently interested in automatic tappets for the manufacturers of these units to maintain a commercially worth-while level of production. The tappets operate the valves through tubular push-rods and rockers in the orthodox manner, adjustment being provided in the push-rod.

Lubrication is provided by a gear-type pump with screened pickup which is driven from the front end of the crankshaft, the spindle being carried through to drive the distributor, which is mounted on the top side of the engine. The oil pump delivers via the main delivery pipe to a full-flow oil filter unit, and from this filter unit to the main oil passages from where oil is fed to the crankshaft and camshaft bearings and to oil jets, which direct oil onto the timing chain and crankshaft skew gears. The oil filter is arranged with a bypass valve, in the normal manner.

The cooling system is arranged with the radiator behind the engine. The coolant pump is mounted above the crankshaft centerline and is driven from the front end of the crankshaft by a V-belt in the form of a triangulated drive, which incorporates the generator, which is mounted toward the left

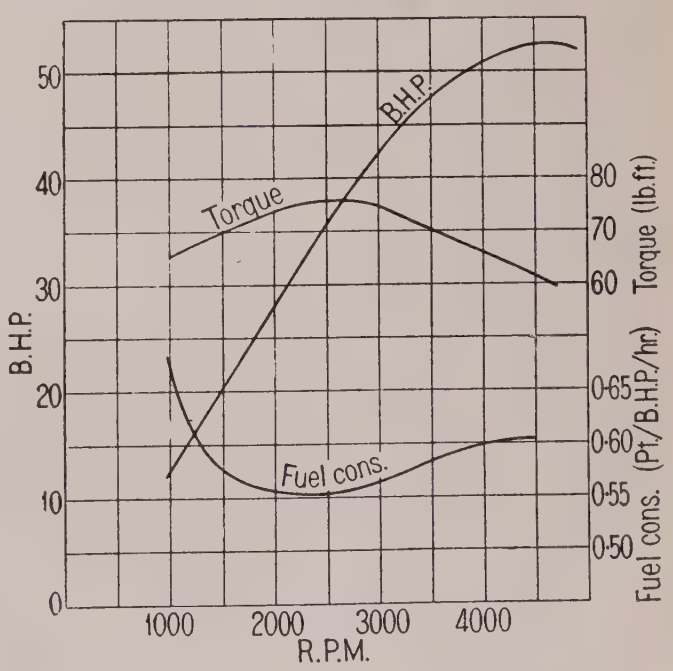

Fig. 14 – Performance curves for Jowett Javelin engine

side of the engine. At the opposite end of the coolant pump spindle, in front of and adjacent to the radiator, is the fan, which blows air through the radiator, the spindle being mounted on a ball bearing to take the belt load and two Reservoil plain bushings.

With 7.2/1 compression ratio on 70-octane-number insensitive gasoline, the engine (Fig. 14) gives 52.5 hp at 4500 rpm and 76 ft-lb of torque at 2600 rpm, a maximum bmep of 126 psi. Minimum fuel consumption of 0.55 pt per bhp-hr is recorded between 2200 and 2500 rpm.

Jaguar XK 6-Cyl Engine

Figs. 15 and 16 show the 210 cu in. Jaguar XK 6-cyl passenger-car engine made by Jaguar Cars Ltd., Coventry.

This engine (of 3.27-in. bore x 4.17-in. stroke) was designed and developed in the immediate post-war period, for the Jaguar XK120 and 120C 2-seater sports and competition models, and the unit is also used in the Jaguar Mark VII 6-seater sedan car of 3700 lb gross weight.

The Jaguar company has always been interested in marketing high-performance cars in the medium-price bracket. They are comfortable, tractable, reliable, and generally domestically usable vehicles.

In choosing the double overhead camshaft engine with hemispherical head, the engineers responsible can be said to have made a bold but logical solution to their engine problem. When the engine was first introduced there was some doubt in certain quarters as to whether such a power unit would prove satisfactory for the work-a-day car. General customer acceptance and satisfaction appear to have amply justified the designer's conception of the type of engine he required for his car.

The engine, while not being designed as a racing engine in any way, has been highly successful, especially with performance-increasing modifications in international trials and races since its introduction.

In general, the engine layout can be said to be fairly orthodox for a 6-cyl double overhead camshaft in-line engine, but many of the detail design points, especially in the layout of the cylinder head and porting, show advanced design and some original features.

In his recent paper on the Jaguar engine (read before the Institution of Mechanical Engineers in England), W. M. Heymes, chief engineer of the Jaguar company, said that in his view the advantages of the hemispherical head over other types are so many that when carefully analyzed he couldn't understand why this type of head is not more generally employed. He listed these advantages:

1. Valve throat flow is probably as good as can be obtained, owing to the blending surfaces of the hemisphere, permitting the whole valve to be effective on both intake and exhaust.
2. Turbulence is easily controlled by port shape to almost any extent.
3. Exhaust-valve cooling. A good flow of water to the seat is obtained, due to the open angle of the head surface with the valve inside the water jacket.
4. Machining. This is the simplest of all types. A single form cutter is used on a fixed center, owing to the fully symmetrical shape.
5. Service facility is good. The head can be removed complete for attention to valves, and there is complete freedom from interim valve adjustment.
6. Burning can be ideal when turbulence is correct and with a central spark plug.

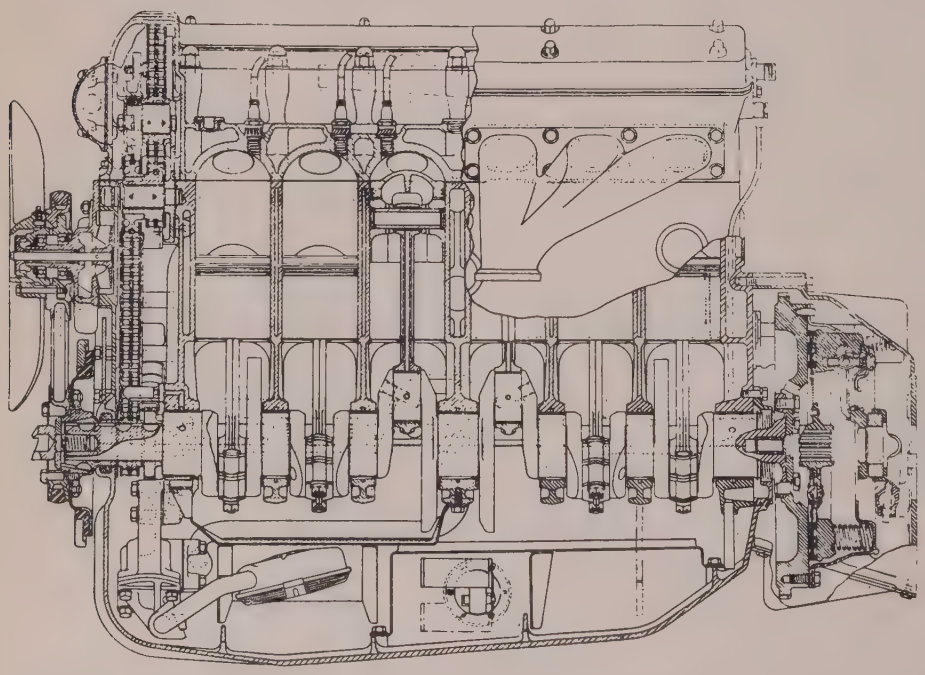

Fig. 15 – Jaguar XK 6-cyl engine

7. Self-ignition or running-on is practically nonexistent.

In this design the valves are installed with an included angle of 70 deg, being evenly split on either side of the cylinder axis. The intake valves are made of silchrome I steel having a throat diameter of 1.375 in. The exhaust valves are austenitic steel, unfaced, with a throat diameter of 1.25 in. Each valve has a pushed-in interference fit seat ring in high-nickel austenitic cast iron. These rings are fitted with the cylinder head heated to 150 C with the ring at room temperature. The valve lift on the standard engine is 0.3125 in. for both valves, with valve timing:

	Mark VII	XK 120
Inlet:	Opens 10 deg early	15 deg early
	Closes 50 deg late	57 deg late
Exhaust:	Opens 57 deg early	57 deg early
	Closes 15 deg late	15 deg late

Much attention was given, in the design and development stage, to the valve port layout. The intake port is offset from the valve seat and curved to give a directional bias to the induced charge, which in turn gives a swirl turbulence in the combustion chamber, the swirl being directed toward the spark plug. It is claimed that this principle of directional swirl turbulence into a hemispherical combustion chamber reduces the octane requirement of the engine well below what it would be if the port was straight in line with the intake valves.

Much development attention was given to flow through the ports at increasing increments of valve lift, the endeavor being to obtain the maximum flow by arriving at the best possible port and valve head form so as to take full advantage of the design.

The exhaust system is split in two sets of three-branch manifolds; a development to give the best engine scavenging throughout the speed range.

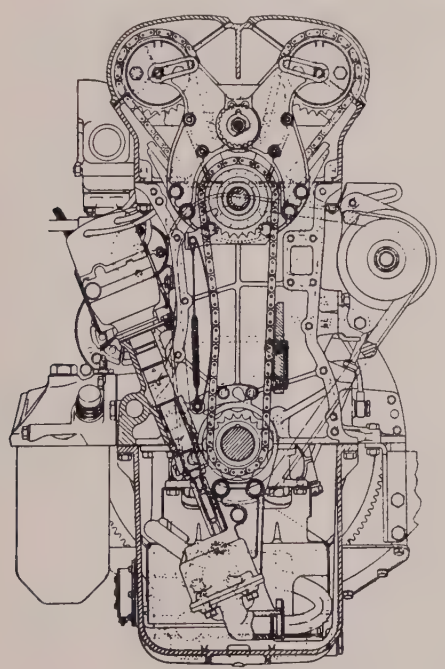

Fig. 16 – Jaguar XK 6-cyl engine

The main structural member of the engine, the cylinder block and crankcase unit, is cast iron, carrying six plain cast-in cylinder bores, each of which is separated from its neighbor and completely surrounded by coolant. This helps to reduce distortion.

The cylinder head of the engine is aluminum alloy, mainly to save weight, as a cast-iron head of similar design would be some 70 lb heavier.

The engine sump is a deep aluminum-alloy casting of rigid structure and has been carefully designed with baffles to maintain a continually submerged oil pump inlet under conditions of violent acceleration and deceleration, which a high-performance car is subjected to.

The cylinder block with this type of engine can be, and is in this engine, a fairly simple casting, the weight of which has been kept down by terminating the crankcase sides level with the crankshaft axis and having a sump mounting flange of very ample proportions to stiffen the structure in lateral bending. This is claimed as a very definite design feature of the engine. Further stiffness is added to the crankcase by the walls, which carry the seven main crankshaft bearings, and particular attention has been given to tying up the main bearings and the cylinder head. Webs from the cylinder-head studs traverse the water jackets to form a tie with the main bearing studs.

The 7-bearing crankshaft has main bearings of 2¾-in. diameter with 23/32-in. pins. The shaft is partially counterbalanced, the scheme being to relieve the load on the center main bearing without adding too much weight to the shaft by fully counterbalancing, as it was realized in the design stage that crankshaft mass in this high-speed engine would be a problem from a torsional point of view unless this mass was kept to a minimum, accepting the fact that it is easier to deal with bearing loads within limits than with torsional vibration problems.

A torsional vibration damper is mounted externally on the front end of the crankshaft. This damper is formed on a steel mounting flange attached to the crankshaft bonded to which is a thick rubber ring, which carries, bonded to itself, a malleable iron damper ring.

The crankshaft is heat-treated prior to machining to give a Brinell figure of 270 to 295, and is carried on seven precision-fit white-metal-lined steel-backed shell bearings; for competition work indium flash lead-bronze bearings are used with some increase in diametrical clearance. The crankshaft pins and journals are finished by handlapping to 10 micro-in.

Oil is fed to the big end bearings from the mains via diagonal feed holes, which terminate at the axis of the drilled crankpins, the crankpins being hollow drilled at an angle to form a sludge trap and a

transverse hole is drilled through the crankpin to feed oil to the big end bearings 90 deg in advance and retard of the maximum radius point.

The camshaft and accessory drives are all taken from the front end of the crankshaft.

A gear oil pump and distributor is driven by skew gears, the overhead camshafts by duplex roller chains via a 2-stage system with tensioners and dampers, which system has been the result of much development work to reduce noise. A triangulated V-belt drive operates the generator, fan, and coolant pump.

In choosing double overhead camshafts as the best method of operating the valves on this engine, bearing in mind its wide speed range, the designer claims the following advantages for such an arrangement:

1. The low reciprocating weight of the valve parts permits the valve spring strength to be approximately half of that required for a crankcase-mounted camshaft, push-rod, and rocker arrangement. This means that one of the chief causes of valve seat wear and valve breakage is eliminated as are also the high tappet loads that occur with a crankcase-mounted camshaft arrangement.

2. The absence of wear between the tappet and the valve by the virtual elimination of side movement across the valve tip, which exists with a rocker arrangement, setting up thrust on the valve stem.

3. The protection that the inverted tappet gives against excessive oil flow down the valve guides.

The initial adjustment of valve clearance is a precision job and is done by first using a set minimum shim thickness in all tappets, installing the camshaft and noting the clearances, then removing the camshaft and installing the right thickness of shim, which is retained in the tappet. This job is done with the head on the bench. When done the clearances stay put.

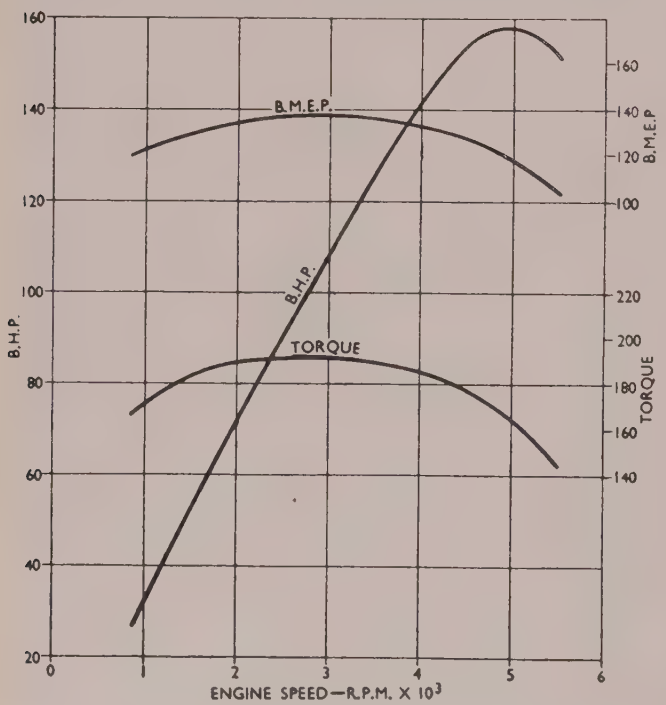

Fig. 17 – Performance curves for Jaguar engine – 8/1 compression ratio

The engine cooling system is pressurized to 4 psi by means of a spring-loaded radiator filler cap. The centrifugal pump draws coolant from the radiator and forces it through a cast-in manifold in the cylinder block, which runs the full length of the cylinder block on the exhaust side, coolant being circulated from the block into the cylinder head first around the exhaust port and valve seat, then over the top of the combustion chamber around the spark-plug boss, over the intake valve port, and into the jacket of the water-heated induction manifold, then through the thermostat to the radiator, the cylinder barrels being cooled by thermosiphonic action.

Performance Figures – The standard engine (Fig. 17) with 8/1 compression ratio on 80-octane (Motor Method), 90-octane (Research) gasoline, fitted with 2 SU H6 1.75-in. bore carburetors, gives the following:

1. At 2500 rpm, 93.5 bhp, 195 ft-lb torque, and 140 psi bmep.
2. At 5000 rpm, 158 bhp, 165 ft-lb torque, and 118 psi bmep.

This engine performance is sufficient to propel the Mark VII sedan at a genuine 100 mph.

M.G. T.D. & 'B' Series Engine

The 1¼-liter 4-cyl sports car engine shown in Figs. 18 and 19 is manufactured by the Nuffield Group for its M.G. chassis, the M.G. car being built at Abingdon-on-Thames.

The engine is based on the 10-hp valve-in-head Morris passenger-car engine, but has many modifications and refinements to give high performance and durability.

The basic engine was designed in 1936 and was thus influenced by the then existing horsepower tax based on piston area.

With a bore of 2.618 in. and stroke of 3.543 in. (giving a capacity of 76.3 cu in.), this engine, with its in-line overhead valves operated from a side camshaft by push-rods, is of orthodox design, which has been stabilized for a number of years.

The major structural members of the engine, the crankcase and cylinder-block unit and cylinder head, are of cast iron, with a deep aluminum-alloy sump.

The engine is produced in two versions, the one for the M.G. Midget Series T.D. sports car, having two 1¼-in. bore sliding choke S. U. carburetors, whilst the other versions for the Series 'B' 4-seater sedan has one carburetor of the same type as the sports car.

In the case of the 2-carburetor installation the semihorizontal S. U. carburetors feed through short induction pipes almost directly to their respective pairs of intake valves for the two cylinders with which each carburetor deals. The single carburetor installation feeds through a Y-type cast-

iron manifold, which is integral with the exhaust manifold, to the induction ports with a static hot spot arranged from the exhaust from the center pair of cylinders.

The inclined valves are set at some 5 deg from the vertical in the cast-iron cylinder head and are arranged with the inlet valves adjacent in the orthodox manner for a 4-cyl engine. The combustion chambers, of the offset bathtub type, give a moderate amount of compression turbulence and have a compression ratio of 7.3/1 for both engines.

Both intake and exhaust valves are of silchrome I steel and are fitted directly into 30-deg seats cut in the cylinder head, without seat rings. The throat diameters are 1.18 in. for the intake, 1.02 for the exhaust, with a maximum valve lift of 0.3125 in. Valve timing is:

Intake: Opens 11 deg early
Closes 57 deg late
Exhaust: Opens 52 deg early
Closes 24 deg late

The cylinder block carries the four integral cylinder bores, which are surrounded by coolant for their full length. Coolant is pumped into the cylinder block by a vane-type pump driven by a V belt from the front end of the crankshaft. This pump circulates coolant through the cylinder block via transfer passages arranged adjacent to the exhaust valves and spark plugs into the cylinder head, the coolant being taken from the front end of the cylinder head, where a thermostat is located.

The forged-steel 3-bearing crankshaft is carried on steel-backed white-metal bearings with end location at the center main bearing. Diagonal oil-feed holes carry oil from the main bearings to the big end bearings, which are also of white metal, steel backed.

The crankshaft bearing dimensions are such that there is some overlap between the main journals and crankpins, and integral counterbalance weights are incorporated to relieve centrifugal loading on the main bearings.

The connecting rods are steel forgings with the piston pin located in the rod by angle-mounted setscrews.

The flywheel is a low-phosphorous iron casting and is secured by two dowels and four setscrews to the rear flange of the crankshaft.

The 3-bearing steel camshaft is driven from the nose of the crankshaft through a duplex roller chain tensioned by a curved shoe, which has raised ribs to bear on the chain rollers. Tension is, in part,

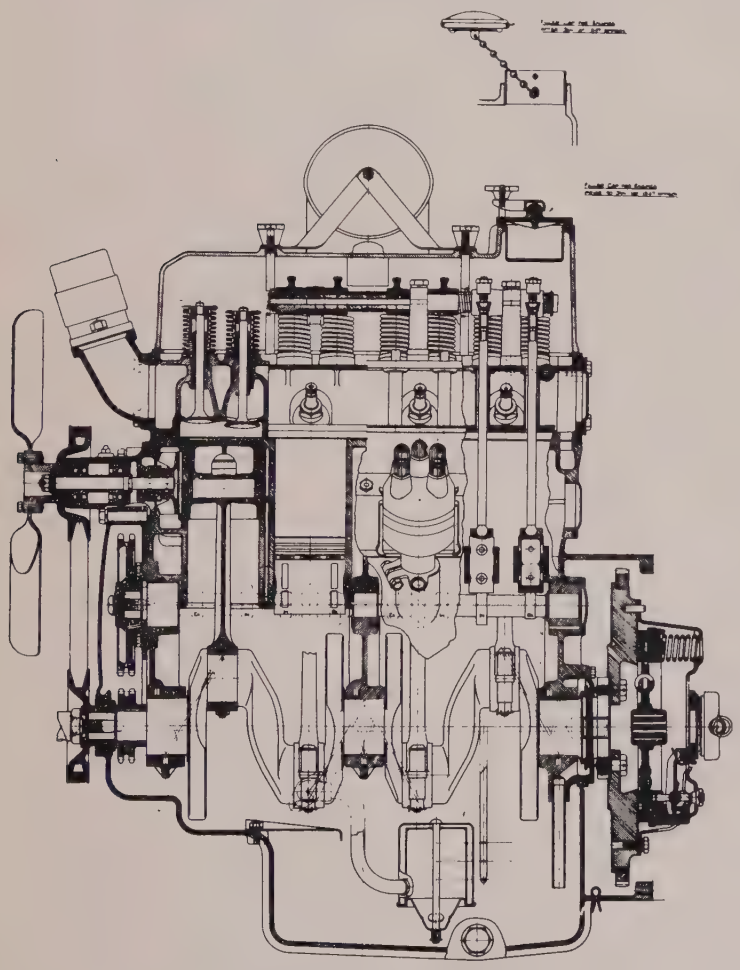

Fig. 18 – M.G. 4-cyl engine

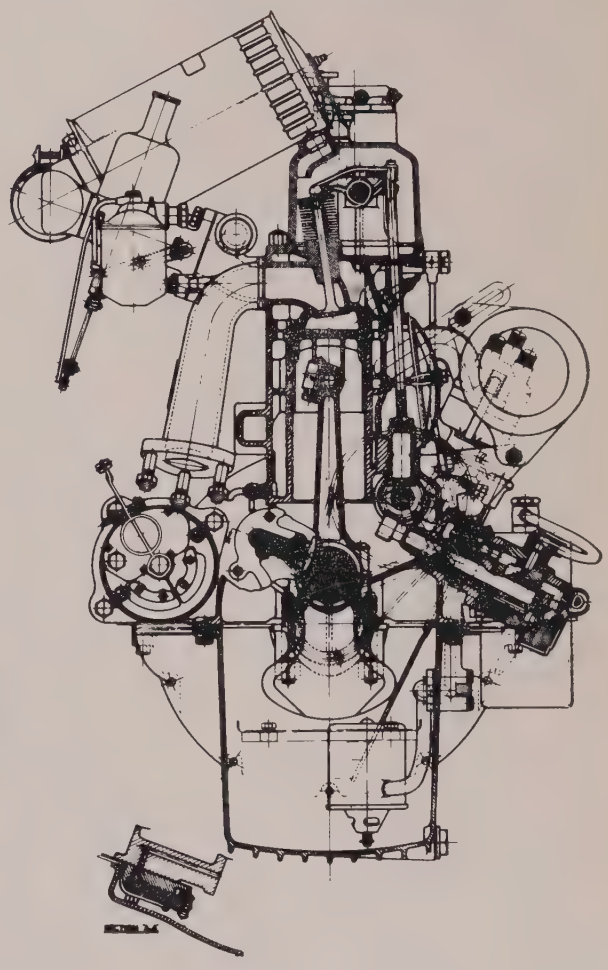

Fig. 19 – M.G. 4-cyl engine

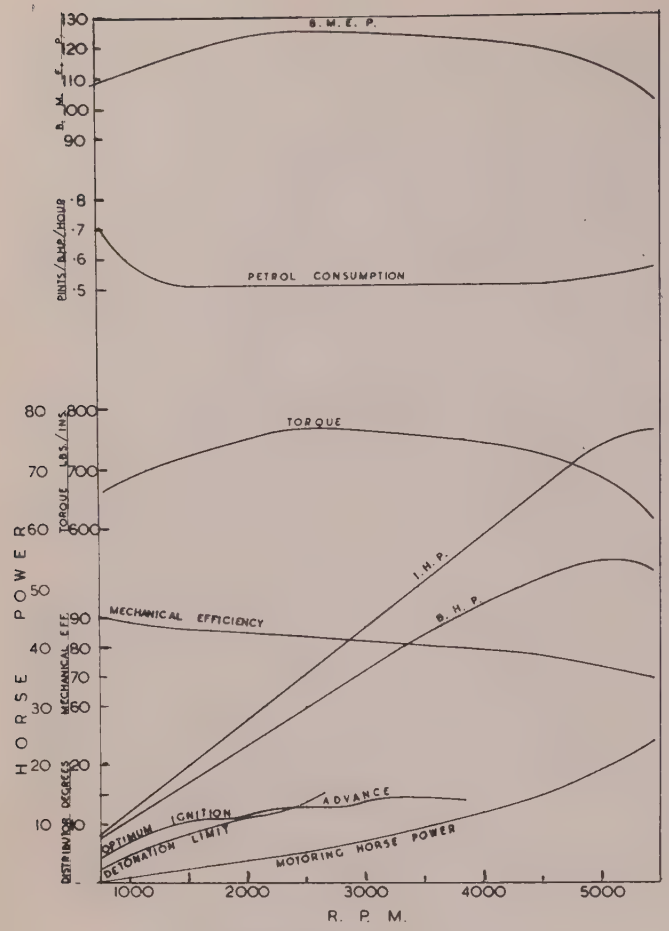

Fig. 20 – Performance curves for M.G. engine

maintained by a light spring backed up by hydraulic pressure from the front main crankshaft bearing.

The camshaft has two skew gears cut on it; one to drive the distributor, the other to drive the externally mounted gear-type oil pump, the externally mounted oil pumps being accepted practice on most Nuffield engines.

The oil is delivered from the oil pump by an external pipe to a full-flow filter bypassed, in case of obstruction, by a ball valve loaded to 60 psi. Oil is fed by a drilled passage in the crankcase to the camshaft and crankshaft main bearings and by an external pipe to the hollow valve rocker tube; oil being returned to the crankcase and valve cover via the push-rod guide tubes. The camshaft operates the valves through barrel-type tappets, tubular push-rods, and rockers, all valves being fitted with double valve springs to allow satisfactory high-speed operation.

One of the main design features of the engine is that the crankcase is split on the centerline of the crankshaft, being provided with a generous flange for mounting the sump. This arrangement is claimed to have the advantages that it lightens the main structural member of the engine. The cylinder-block-crankcase unit has ample stiffness in the horizontal plane, provided by the thick sump flange, while the aluminum-alloy sump is studded to the clutch bell housing, thus making some contribution to the stiffness of the unit as a whole and in maintaining alignment of the crankshaft with the gearbox first motion shaft.

With two carburetors the engine develops (Fig. 20) 54 hp at 5200 rpm with a maximum torque of 63.7 ft-lb at 2600 rpm and a maximum bmep of 125 psi. With a single carburetor the performance is reduced to 46 hp at 4800 rpm and 58.5 ft-lb torque at 2400 rpm being 116 bmep. The firing order of the engine is the orthodox 1-3-4-2. The car weight the engine has to deal with is in the region of 2100 lb for the sports car and 2200 lb for the sedan.

Rolls-Royce and Bentley Mark VI Engine

The 6-cyl in-line Rolls-Royce engine (Figs. 21 and 22) is used in both the Rolls-Royce and Bentley cars. Both cars are manufactured by Rolls-Royce Ltd., at their Crewe works.

The main point of difference is that the Bentley Mark VI engine is fitted with two horizontal S.U. sliding-choke fully automatic carburetors, whilst the Rolls-Royce engine has one fully automatic down-draft Zenith carburetor.

The choice of an in-line 6-cyl engine for these cars is, in some way, influenced by the company's policy to achieve a sleek and somewhat greyhound like appearance of the front end of the car, realizing that the Rolls-Royce and Bentley radiators have a traditional appeal to the customer.

This in-line engine has also made possible a rationalized range of engines of 4, 6, and 8 cylinders, using a large number of common parts. This policy has been of considerable benefit to the British fighting services, since the Rolls-Royce 4-, 6-, and 8-cyl 'B' series engines for the military are very similar to the car engines. Had the engine, as a basic type, been of V design, a rationalization policy would have been very difficult because very little can be done with the V engine when it comes to common parts. If the number of cylinders is to be varied to produce a range of engines, the angle between the V alters with the number of cylinders and thus there could not be many common parts. If the angle is kept the same with a varying number of cylinders, then unequal firing orders have to be accepted on all but one of the combinations, and this would seem to be wrong.

This in-line engine is of F-head design, the major structure of the engine, the cylinder block and crankcase, being of unit construction in cast iron; the cylinder head, sump, and other parts being made mainly of light alloy.

The aluminum cylinder head carries the silchrome I intake valves, which seat onto screwed-in steel seat rings. The intake valve is of 1.75-in. throat diameter, 0.343-in. stem diameter, and has 0.400-in. maximum lift. The austenitic stellite-faced exhaust valves in the iron cylinder block seat directly onto the block deck without seat rings, and have a throat diameter of 1.5 in. with 0.373-in. stem

and 0.375-in. maximum lift. The valve timing is:

Inlet: Opens 20 deg early
Closes 90 deg late
Exhaust: Opens 58 deg early
Closes 33 deg late

This valve timing was established by single-cylinder and full-scale engine tests where, in the single cylinder engine, adjustments to valve duration, phasing, and lift could be quickly and conveniently carried out during development of the design.

The F-head arrangement was chosen after considerable testing of alternative designs, including overhead camshaft arrangements with wide-angle valves. The chosen valve arrangements allow a large-diameter intake valve and sufficient lift to give good breathing without mechanical embarrassment such as interference between the valve and the piston. Overhead camshaft designs, which gave good power, were unacceptable for noise level and general valve gear and camshaft drive rattle. It has always been Rolls-Royce policy to make the engine as unobtrusive as possible to the passengers.

The stroke/bore ratio of the engine was in some ways a concession to the national automobile tax system in the U. K., now abandoned, when the tax on a car was charged on total piston area.

The present 278 cu in. engine, with its 3.625-in. bore and 4.5-in. stroke is considered of reasonable proportions. Life tests have confirmed this.

A considerable amount of money is spent on the crankshaft and rifle-drilled connecting rods. These are machined all over from steel forgings. This is worth while because this engine has the smallest piston pin, crankpin, and connecting-rod bolts in the industry for its size, the piston pin being 0.750-in. outside diameter, bored out 0.375 in., the big end journal being 2-in. outside diameter, and the rod bolts 0.343-in. outside diameter; internally bored to spread tensile stress as evenly as possible over the whole bolt. The connecting-rod length of 7.950 in. centers gives an L/S ratio of 1.765.

The crankshaft has seven main bearings of the precision-fit thinwall indium-copper-lead type, and is fitted with bolt-on balance weights and has bored out pins and journals of 380 Brinell surface hardness. Oil is introduced to the big end bearings, which are of similar type to the main bearings, 90 deg in advance and trailing of the maximum radius point of the crankpin. The oil holes in the pin are so arranged as to prevent the big end bearings from being damaged by sludge, which is trapped in a sludge trap formed in the crankpin.

The helical timing gears are at the front end of the engine, for simplicity and accessibility. The virtue of placing these gears at the rear of the engine is far outweighed by the problems encountered of fixing the flywheel and overhang of this major mass. Further, with rear-mounted timing gears, noise problems arise inside the car. The servicing of rear-mounted timing gears means removal of the flywheel, which is a job to be avoided if possible. The camshaft gear is made of RR.50 aluminum alloy. Experiments with plastic fabric gears have shown the life of these to be dubious. Moreover, no

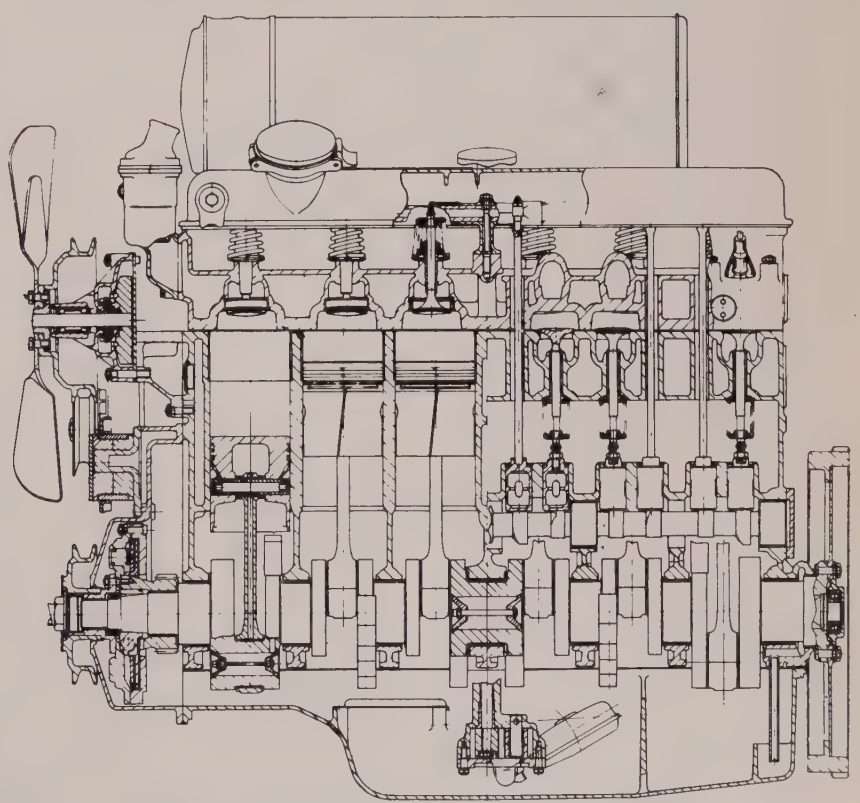

Fig. 21 – Bentley 6-cyl engine

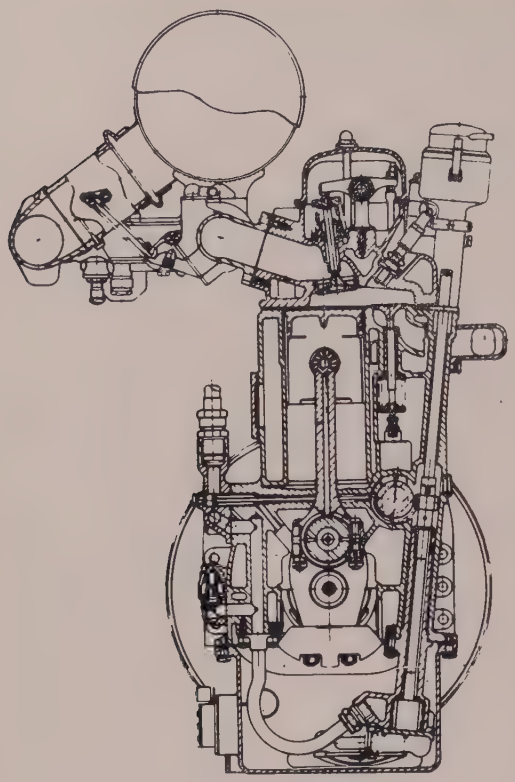

Fig. 22 – Bentley 6-cyl engine

sure way has been found of physically inspecting such gears for soundness of material of construction. The RR.50 gears have shown themselves to have an adequate safety factor.

An internally mounted fully lubricated slipper-type Lanchester damper is fitted to the nose of the crankshaft. This damper reduced torsional vibration periods, except the third-order period, to under ± ½ deg. The third-order period peaks at 5050 crankshaft rpm, which is outside the maximum permitted speed of the engine. To safeguard the camshaft wheel when running with an amplitude of ± 1.4 deg, the steel crankshaft pinion is spring-driven and capable of an angulated movement of ± 5 deg at the rate of 60 lb-in. per deg.

All Rolls-Royce and Bentley engines receive some considerable running in before leaving the Crewe works. Each engine is run 5 hr on the dynamometer, followed by a 100-mile road test in its car, when oil consumption is checked by weighing.

Rover 75 Engine

The Rover 75, 6-cyl in-line engine (Figs. 23 and 24) is manufactured by the Rover Co., Ltd., Solihull, Birmingham, the series No. 75 being an indication of the maximum bhp of the engine in its standard production form. This engine is designed for a high-quality family-type sedan of around 3200 lb gross weight.

The layout of the top part of the engine can be said to be definitely unorthodox, the design being a somewhat distorted F-head layout with the combustion chamber designed around the spark plug and formed mainly in the cylinder block.

A 6-cyl in-line engine was chosen for the Rover car because it was not considered possible to achieve the required degree of refinement of performance necessary for this type of car with a 4-cyl unit. However, a 4-cyl version of this unit is manufactured for the Land Rover car, which is a "jeep" type of vehicle, sold mainly for agricultural and military purposes.

Unconventional cylinder layouts were avoided, as it was considered that the unorthodox layout of the top end of the engine in the form of combustion chamber and valve gear was sufficiently new to occupy the development resources of the company for one model.

The main structural member of the engine, the cylinder block and crankcase unit, is made of cast iron, the cylinder head being aluminum alloy. Push-fit cast-iron inserts are provided for both intake and exhaust valves.

The bore and stroke are 2.567 x 4.134 in., giving a stroke/bore ratio of 1.61/1. These dimensions, when the engine was designed in 1938, were primarily dictated by the now obsolete taxation system based on total piston area. At the same time it must be said that the choice of cylinder dimensions was also influenced by the desire to reduce engine length to a minimum.

The choice of F-head layout using inclined valves arose from the desire to obtain high thermal and volumetric efficiencies, combined with mechanical simplicity.

For thermal efficiency this combustion chamber provides adequate compression turbulence, short flame travel, and a spark plug located near the exhaust valve. This spark plug is scavenged by charge movement during compression. It is claimed that, with the compact combustion chamber formed, the engine has the ability to operate at high compression ratios and to burn very lean mixtures at part throttle on fuels of very limited octane number.

For volumetric efficiency this form of combustion chamber allows an intake valve of almost unrestricted diameter located directly over the cylinder bore, with an exhaust valve located in such a manner that it is not swept by the incoming charge, yet providing a direct exit for the exhaust gas.

For mechanical simplicity all valves are operated by a single camshaft located fairly high in the cylinder block and driven by a duplex roller chain from the front end of the crankshaft.

The exhaust valves are operated directly from the camshaft by rocker-type cam followers. Similar cam followers operate the intake valves via push-rods and overhead rockers. The silchrome I intake valves are of 1.562-in. throat diameter with a lift of 0.280 in. The XB exhaust valves are Brightray faced of 1.125-in. throat diameter with a lift of

0.375 in. These unconventional dimensions are, in the case of the intake valves, to keep down the stress in the push-rod valve gear, the low lift being compensated for by the large diameter, which allows sufficient valve annulus area for the engine.

A large-diameter exhaust valve would cause a higher valve temperature through greater surface area, and therefore, a large and hotter spot in the combustion chamber. For this reason the valve is kept small and full advantage is taken of its low mass and the great stiffness of the exhaust-valve gear train to use a high lift. This in turn gives the full area of the exhaust-valve throat over a considerable portion of the lift curve.

The valve timing is:

Inlet: Opens 10 deg early
Closes 40 deg late
Exhaust: Opens 42 deg early
Closes 12 deg late

In considering the design requirements, the following points are made for not adopting alternate types of combustion chamber.

Side-Valve Types – Whilst these have adequate compression turbulence, breathing is very restricted, due to the limitations in valve size and tortuous entry and exit paths to the cylinder. Further, the combustion chamber is not sufficiently compact, and cooling around the exhaust valve is difficult.

Overhead-Valve In-line Types – While better than the side-valve type for breathing, its valve size is limited by engine length unless the cylinder proportions are nearly square. Cooling between inlet and exhaust valves is not good and high thermal stress occurs between the valve seats. Plug position is thought to be worse than on a side-valve engine, in respect to flame travel, and compression turbulence is barely adequate.

Overhead-Valve Double-Row Types – There are many varieties of such layouts, of which the hemispherical layout is the best known. Whilst volumetric efficiency is excellent with the hemispherical layout, it is thought that the low-speed and light-load performance of this layout is impaired by lack of compression turbulence and the plug position is little better than that on the valve in-line types if adequate valve sizes are used. This results in rather long flame travel or a heavily masked plug, giving poor ignitability of lean part-throttle mixtures.

Other types have shaped piston crowns and compact, offset combustion chambers with compression turbulence and a plug located at the side near the exhaust valve. They are much more attractive, but they still lack the compactness and good plug scavenging of the inclined-valve F-head type. In addition, without using an overhead camshaft, valve operation is difficult to achieve, and an overhead camshaft can be costly. This makes top overhauls more complicated and its main problem of drive is difficult to obtain with satisfactory silence.

F-Head Types with Vertical or Nearly Vertical Valves – The combustion chambers of these designs appear to lack compactness, especially if the exhaust port is positioned to permit adequate cooling between it and the cylinder barrel. Where a step combustion-chamber roof is used in preference to a shaped piston crown, the scavenging of the plug may be impaired. Flow past the exhaust valve can also be restricted with this type in that the exhaust valve has the shortcomings of an orthodox L-head engine.

Horizontal Valve Types – These, whether of the opposed-valve or parallel-valve types, suffer from somewhat restricted gas flow and valve installation is difficult. The whole arrangement looks complicated mechanically, with nothing to recommend it.

The camshaft is driven by a duplex chain. Chain

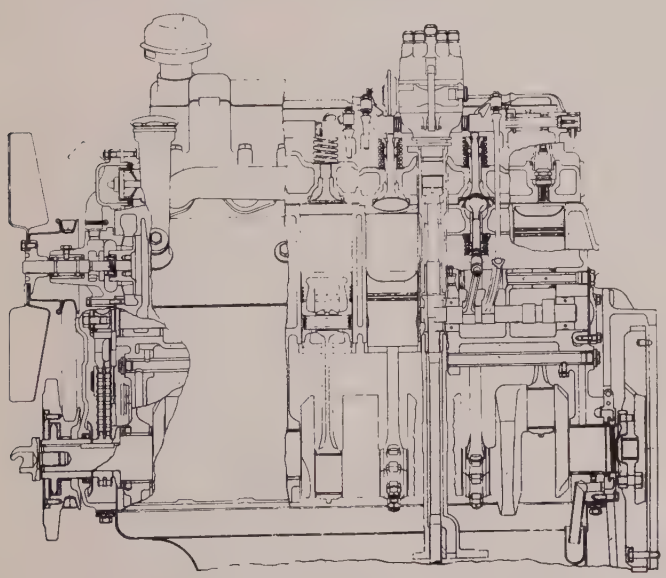

Fig. 23 – Rover 6-cyl engine

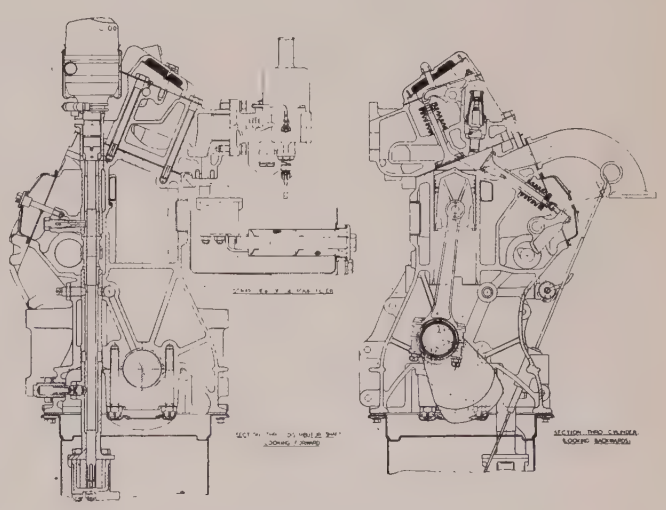

Fig. 24 – Rover 6-cyl engine

drive was used for the following reasons:

(a) The drive had to be common with that on a 4-cyl engine, where gear drive had proved excessively noisy.

(b) The centers were such that an idler gear was necessary, which made gear drive still less attractive.

A spring-loaded sprocket-type chain tensioner is used with one-way hydraulic damping.

The ignition distributor and oil pump are driven by skew gears from the camshaft, the distributor being mounted above the cylinder head on the exhaust side of the engine, where it is adjacent to the spark plugs and relatively immune from splash when fording. This distributor position has proved a very useful asset on 4-cyl Land Rover engines. The water pump fan and generator are belt driven by a V belt from the front end of the crankshaft, the pulley for which is formed integrally with the crankshaft torsional rubber-mounted damper.

The crankshaft is a steel forging having four main bearings all of 2.25-in. journal diameter, the crankpins being 1.875-in. diameter. The crankshaft is provided with integral mass balance to relieve the load on the main bearings.

The connecting rods are steel forgings, the piston pin being 0.6875-in. diameter hollow parallel bored 0.437-in. full floating and secured in the aluminum piston by circlips.

The aluminum tin-plated pistons are of the horizontal slotted-skirt type with two compression and two oil control rings, all above the piston pin. There are three horizontal slots on each skirt face. The topmost of these slots is to give thermal control of piston shape. The lower slots are to provide better oil control by preventing oil pumping when the piston thrust changes from one side of the bore to the other.

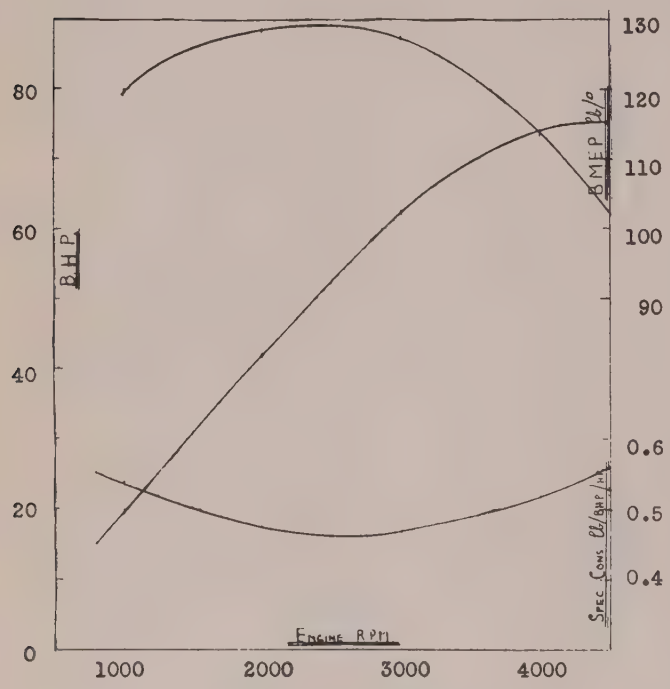

Fig. 25 – Performance curves for Rover engine

The induction system consists of two manifolds, each with its own S.U. fully automatic sliding choke carburetor feeding three cylinders. The manifolds are jacketed and cast integrally with the aluminum cylinder head, being connected with a 0.375-in. bore balance pipe.

Particular attention was paid in the development stage to obtain good distribution with a cold manifold. These manifolds were developed on a special cylinder head, which allowed the manifolds to be bolted onto the cylinder head.

Forty different designs were tested, made up in wood, which enabled modifications to be made very quickly. It is thought that the distribution pattern is determined by the internal shape of the manifold and does not rely upon heat. Thus, with proper manifold shape giving good distribution, cold, the warmup start and drive away is good.

Manifold heat is provided by the water jacket. One advantage of water heating as opposed to exhaust heating is that where engine speed is falling on full throttle, such as hill climbing, the hot-spot temperature does not rise excessively. Two carburetors were used to give increased torque in the midspeed range between 1500 and 3000 rpm. The gain in torque with two carburetors between 2000 and 2500 rpm is about 5% over that obtained with a single carburetor.

The S.U. constant-vacuum carburetors are used because, for a given low-speed performance, they give better filling at high speeds than do the fixed choke types. Also, because only one jet assembly per carburetor is used, there is no trouble with rich or weak spots in the carburetion, such as can occur when changing from idling jet to main jet on a fixed choke carburetor. Because these carburetors give a much richer mixture under pulsating airflow than under steady airflow conditions, the required enrichment from most economical cruising mixture between 17 and 19/1 air/fuel ratio to full-power mixture between 12.5 and 13.5/1 air/fuel ratio is obtained by adjusting the size of the balance pipe between the manifolds.

Until the advent of premium gasoline in the U. K. in April, 1953, the fuel available was a 70 octane (Motor Method) fuel of very low sensitivity. This had, in production, limited the choice of compression ratio to 7.25/1. The fuel quality situation in world export markets has, the Rover company feels, not become sufficiently clear to allow further rise of compression ratio to take advantage of higher octane fuels.

The engine gives its maximum torque of 111 ft-lb at 2400 rpm, being 129 bmep and 50 bhp at this speed. Maximum power, 75 hp, is obtained at 4250 rpm, the bmep being 108 psi. Minimum specific fuel consumption at maximum torque is 0.46 lb

per bhp-hr and at maximum power 0.56 lb per bhp-hr. (See Fig. 25.)

Standard Vanguard Engine

The Standard Vanguard passenger-car engine (Figs. 26 and 27) is manufactured by the Standard Motor Co., Ltd., Coventry, for their Vanguard and Triumph cars. Modified versions of this engine, the major parts of which are produced on the same machines, are used in the British Ferguson tractor which is also manufactured by Standard.

The engine was designed as a completely new unit in the immediate postwar period, initially as a power unit for the Standard Vanguard sedan of 2700 lb gross weight. This car was planned primarily on a one-model policy to carry six people and luggage in comfort but to fit into a 14-ft long garage, which is the size of garage prevailing in most domestic dwellings in the U. K. and in many of the export markets.

The engine dimensions chosen (3.347-in. bore x 3.622-in. stroke) give a capacity of 127.6 cu in. They indicate that no attention was paid to the gross piston area tax formula, which was in existence at the time when the design was going through. The main reason for neglecting the formula was that the car was mainly intended for the export market, where such a taxation system did not exist, and to obtain the advantages of cylinder proportions approaching unity.

The engine is designed for ease of production and maintenance and the feature to be most commented on is the use of wet liners, which make for a simple cylinder block casting.

These cast-iron liners are held in compression by the load of the cylinder-head studs via the head face and gasket. The cylinder liners are spigot mounted into 3.625-in. bores in the cylinder-block deck with a flange two-thirds of the way down the liner, which rests on the spigot face. There is a 1/64-in. sealing washer between the liner flange and the spigot face. The top two-thirds of the liner is in contact with the coolant and is held in compression by the cylinder-head stud load with the bottom one-third in the spigot, being free to slide. The liners are positively located to one another by mating flats machined on the top flanges and protrude some 0.004 in. above the top face of the block, being pulled down on the sealing faces between the bottom gasket and cylinder-head gasket.

The cylinder block and crankcase unit extends 3½ in. below the crankshaft center line. It is amply ribbed in between the cylinders and at the three main bearing supports, as it receives little or no reinforcement from the separate cylinder liners. The main bearing caps are registered in the crankcase by machined faces, each cap being retained by two 9/16-in. bolts. The rear main bearing cap forms the end closure for the crankcase, being extended downward to form part of the sump joint face, an oil seal being obtained with a felt-packed vertical slot on either side of the cap. Each main bearing has an annular oil groove for passing oil through radial drilled holes into precision-fit white-metal-lined main bearing shells.

The forged manganese-molybdenum steel 3-bearing crankshaft has integral balance weights, 2.479-

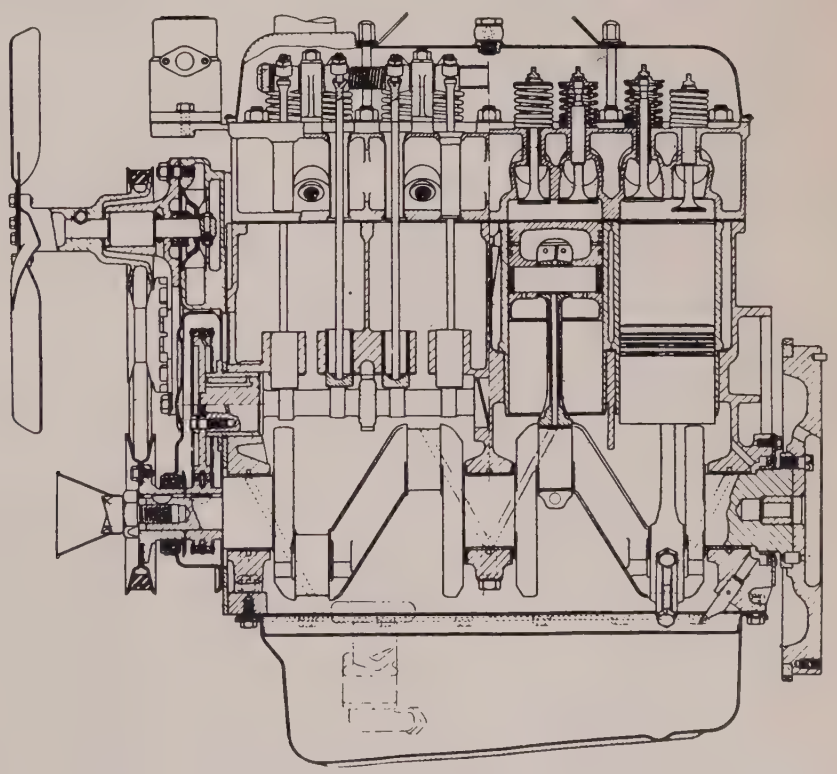

Fig. 26 – Standard Vanguard engine

in. diameter main journals, and 2.087-in. pins, all finished to between 15 and 20 micro-in. The big end bearings are also of the precision-fit white-metal type, steel backed. The main bearings are 1½ in. wide with circumferential oil grooves fed by the four radial holes in order to ensure an adequate flow of oil to the big end bearings, which are fed through diagonally drilled holes in the crankshaft, the exit to these being at the maximum radius point of the crankthrow. Crankshaft end thrust is taken on the center main bearing.

The same material used in the crankshaft is used in the connecting rods, that is, manganese-molybdenum steel. These rods are of H section, with 6.25-in. centers, the big ends are split at an angle of 40 deg from the vertical for ease of assembly and servicing. Bolts ⅜ in. in diameter, locked by a dual tab washer, hold the two halves of the big end boss together. The rods are rifle drilled with a 3/16-in. oilway to feed oil to the piston pin. There is also a cross hole of 1/32-in. diameter located 3½ in. from the big end center, providing a jet to the cylinder walls. The small end bushing in the connecting rod is of the rolled phosphor-bronze type. The piston pin has an outside diameter of 0.875 in. and a bore of 0.578 in., being located by circlips in the piston.

Aluminum-alloy split skirt pistons carry four rings all above the piston pin, the top two of which are compression rings, the bottom two being main and auxiliary oil control rings.

The cast-iron cylinder head is secured to the crankcase by ten 7/16-in. diameter studs with a copper and asbestos gasket interposed.

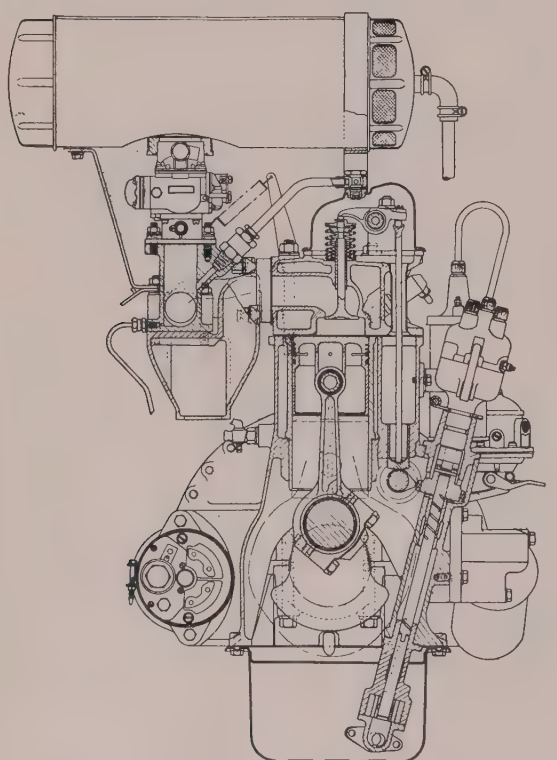

Fig. 27 – Standard Vanguard engine

The combustion chambers are of the offset bath-tub type, providing around 30% compression turbulence over the side of the flat top piston remote from the spark plug, the edge of the combustion chamber remote from the spark plug being relieved to promote flow of charge into the combustion chamber from the squish area.

The cylinder head has all separate ports and the valves are arranged in line in the orthodox manner. The throat diameter of the intake valve is 1 13/32 in. and the exhaust valve 1 3/16 in. The valve lift is 0.3437 in. and the valve timing is as follows:

Intake:	Opens 10 deg early
	Closes 50 deg late
Exhaust:	Opens 50 deg early
	Closes 10 deg late

The intake valves are made of silchrome I steel and the exhaust valves are of XB steel.

An unusual feature of the valve gear is that there are two springs per valve, but the valves themselves have only one internal spring to hold the valve on its seat, which is retained by a slotted collar. At the valve tip, a cap embraces the outer spring, which looks after the rocker push-rod and tappet mass. It is claimed that this arrangement ensures silent operation of the valve gear and reduces the rate of valve seat wear because the whole of the spring load to take care of the valve train does not come on the valve stem. Valve clearances are adjusted through the slot in the side of the outer spring cap, and the valve train is permanently loaded by spring pressure from the outer spring.

The valves are operated through rockers of 1.5/1 ratio by solid push-rods and bucket-type tappets from a 4-bearing cast-iron camshaft. The natural position for the fifth bearing, between No. 1 and No. 2 cylinders, is the location for the drive to the ignition distributor and oil pump, which are on the same spindle axis, driven through skew gears. The camshaft is driven by a duplex roller chain, which is lubricated from four slots located in the front face of the camshaft front bearing. A blade-type chain tensioner is provided.

A diaphragm fuel pump driven from an eccentric near the rear end of the camshaft feeds fuel to a Solex 32-mm down-draft carburetor. An aluminum induction pipe with four branches and thermostatically controlled hot spot feeds the four cylinders. This thermostatically controlled hot spot has shown up to considerable advantage in economy during cold weather over the more orthodox boltup arrangement with hot plate.

The lubrication system is fairly orthodox. Eleven pints of oil are carried in a pressed-steel sump, from which a Hobourn-Eaton (Eaton-Gerotor) pump picks up the oil through a screen suction filter. Oil is passed through drilled oilways in the

crankcase to a bypass filter bolted to the face of the case. The filter passes the oil through to the main oil passage from where it is delivered to the crankshaft and camshaft bearings. Oil fed to the valve gear is taken up from the rear camshaft bearing and controlled by flats on the bearing through drilled holes in the cylinder block and cylinder head to a feed hole in the rear rocker pedestal, whence the oil passes into the center of the rocker shaft and emerges outward at each rocker bearing through a 1/16-in. diameter hole. This system is different on the high-speed Vanguard engine from the low-speed tractor engine, which has a full-flow filter capable of dealing with the full flow in normal circumstances, but there is a bypass device, should the filter become choked. The valve stems have no oil sealing devices apart from the cap that carries the outer valve train spring, which acts as an umbrella to prevent excessive oil passing down the valve stem. With 7/1 compression ratio the engine, with fan, and generator running, and air cleaner fitted, gives 64 bhp at 4000 rpm, a maximum torque of 106 ft-lb at 2400 rpm, with a maximum bmep of 125 psi. These figures are obtained with 70-octane gasoline of low sensitivity. Fuel consumption is rated at full throttle in the order of 53 pt per bhp-hr at between 2000 and 3500 rpm. (See Fig. 28.)

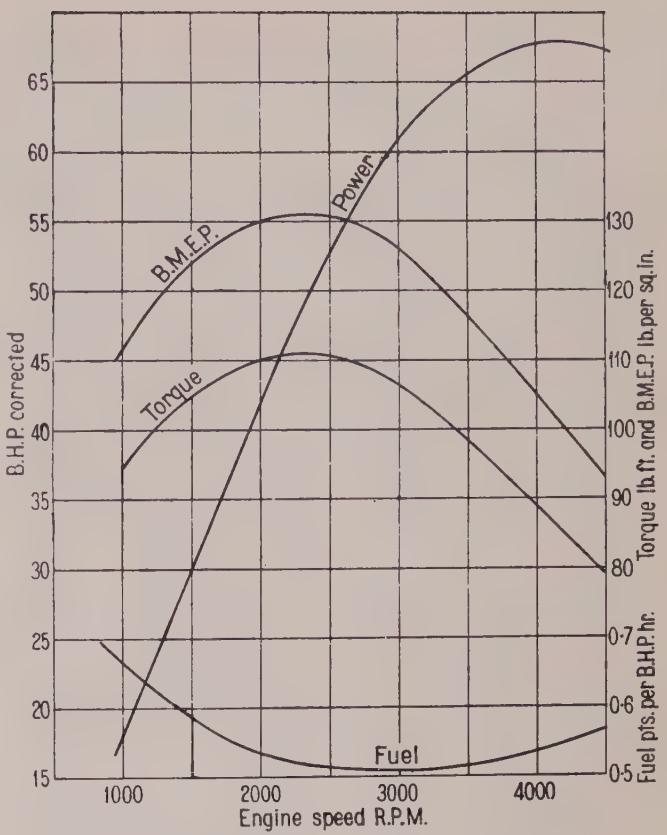

Fig. 28 – Performance curves for Standard Vanguard engine

Vauxhall "Square" Engines

The Vauxhall engines (Figs. 29 and 30) are made by Vauxhall Motors Ltd., Luton, Bedfordshire. There are two power units, which are primarily intended for passenger cars. These units were introduced in March, 1952, being of completely new design and are known as the Vauxhall "square" engines. The two versions of the engine are almost identical in design, except one is a 4-cyl unit and the other a 6-cyl unit.

The 4-cyl unit is used to power the Wyvern model E passenger car of 2205 lb gross weight. This unit is also used in the model CA Bedford light-delivery van, which is also manufactured by Vauxhall. The 6-cyl version of the engine is used in the Vauxhall Velox passenger car of 2313 lb gross weight.

The bore and stroke of 3.125 x 3 in. are common to both engines, giving a swept volume of 92 cu in. for the 4-cyl engine and 138 cu in. for the 6-cyl unit.

The two engines, being very similar in design, utilize the maximum number of common parts for economic production, supply, and general facilities of servicing.

With a known requirement of output and, consequently, of total swept volume, the decision to adopt a stroke/bore ratio of slightly less than unity was in general accordance with several known advantages in favor of square cylinder proportions, and following the change in basic taxation rating in the U. K. there was no longer any need to worry about total piston area from a tax point of view. The prewar Vauxhall engines of relatively long stroke and small bore had been developed under the piston area taxation formula.

A close knowledge of the characteristics of the prewar power units produced by Vauxhall, which preceded these new engines and were of almost identical capacity, enabled a very close comparison to be made and claims for reduced mechanical losses, pumping losses, longer bore, and piston and ring life, together with better operating economy, for the "square" engine to be extensively tested, compared with prewar engines. The new engines proved to be an advance.

Both units are of conventional 4-stroke design, the cylinders being disposed vertically and in line, whilst the intake and exhaust valves are arranged vertically in line in the cylinder head in the orthodox manner, operated by push-rods and rockers from a camshaft located in the crankcase driven by a duplex roller chain from the front end of the crankshaft. Lubrication of this chain is by jet from the main pressure lubrication system impinging on the inner surface of the chain.

The crankshafts of both engines are steel forgings counterweighted to reduce bearing loads, and carried in three precision-fit steel-backed white-metal bearings in the 4-cyl engine, and four in the 6-cyl engine. In view of the stroke/bore ratio of the engines, the crankshafts of both engines are designed with sufficient torsional rigidity to ensure

that no harmonics of the natural torsional frequency are likely to cause roughness or other undesirable effects throughout the operating speed range. Therefore, torsional vibration dampers are not found necessary.

The main and big end bearings are supplied with lubricating oil under pressure from a gear pump submerged within the oil sump and driven at half engine speed through skew gears from the camshaft. Oil is delivered under pressure into a passage running the length of the crankcase and thence by means of cross-drilled holes to main and camshaft bearings. The big end bearings are supplied from grooved main bearings via drilled holes in the crankshaft. A proportion of the lubricating oil is delivered, under pressure, to a replaceable-element bypass oil filter mounted on the left-hand side of the engine. Filtered oil is returned directly to the sump.

The combustion chamber is of the conventional bathtub type designed for compactness, whilst taking advantage of the room made available by the stroke/bore ratio of the engine to install valves and ports of adequate size. The precise shape of the combustion chamber is a result of extensive study of the relative roughness, rate of pressure rise, and bmep as affected by change in the height, plan shape, plug location, and quench area percentage of piston crown coverage. The combustion chamber is offset from the cylinder centerline in order to obtain the desirable quench area, the endeavor of the whole design being to obtain good performance in power and economy with good resistance to detonation.

The compression ratio is moderate at 6.4/1 in order to ensure satisfactory operation on low-grade fuels, which still appear in many areas where these power units are marketed. The degree of success obtained in the design of this chamber is indicated by reference to the bmep, where it will be noted that 120 psi is approached with this ratio. The general structural rigidity of the engine is designed and tested to withstand much higher compression ratios with the attendant higher rates of pressure rise and peak pressures, all without undue increase in engine roughness. Major engine components, therefore, need undergo little change when the compression ratio is stepped up by the general availability of better fuels. The spark plugs are located approximately midway between exhaust and intake valves, installed between the push-rod tubes on the right-hand side, or camshaft side, of the engine.

The ignition distributor driven from the upper end of the lubricating oil pump spindle is also on the right-hand side of the engine.

The intake valve ports in the cylinder head are proportioned to give good gas-flow characteristics through the operating speed range. By virtue of the stepped design of the cylinder head, all ports are very short and exhaust gas heat to water rejected at the exhaust port is minimized, whilst the design also makes for a light-weight cast-iron cylinder head. The intake valves are made of manganese-molybdenum steel and have a head diameter of 1.375 in. The exhaust valves are made of XB steel with a head diameter of 1.25 in. The valves have a common lift of 0.336 in. The intake valve:

Opens 4 deg early
Closes 36 deg late

The exhaust valve:

Opens 36 deg early
Closes 4 deg late

The design of the valve porting and general valve proportioning and timing was undertaken with a requirement of good low-speed torque characteristics rather than maximum power, whilst the power is well maintained up to 4000 rpm.

Each pair of intake ports is siamesed in the cyl-

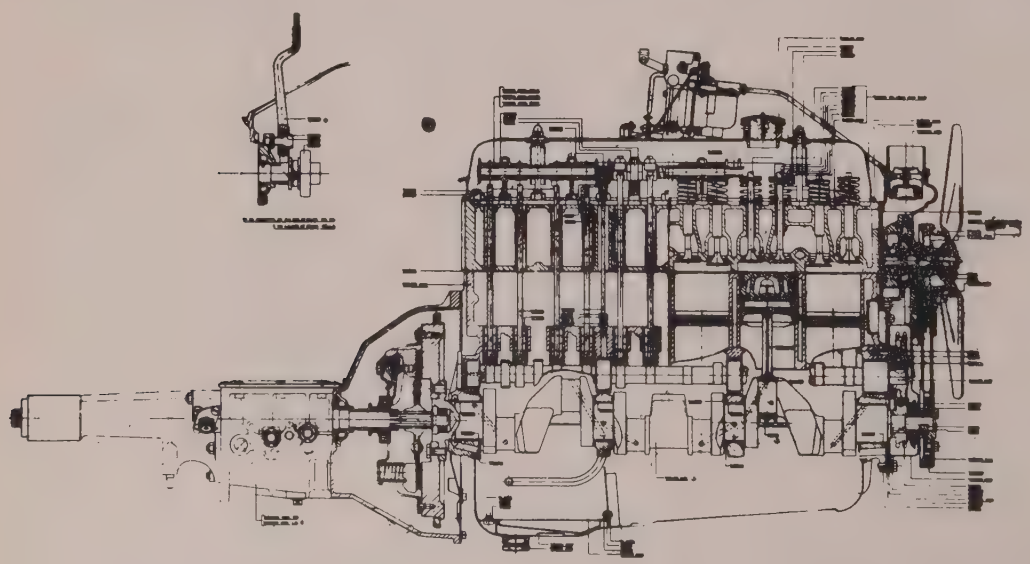

Fig. 29 – Vauxhall "square" engine

inder head and connected to the 30 VIG-7 Zenith carburetor via a 2-port or 3-port cast-iron intake manifold on the 4- and 6-cyl engines, respectively. Immediately beneath the intake manifold and connected to it at the center to form a carburetor hot spot is a cast-iron exhaust manifold connecting with the individual exhaust ports in the cylinder head. No siamesed exhaust ports are employed, thus avoiding a potential source of cylinder-head failure. The supply of exhaust gas to the hot spot under the carburetor is thermostatically controlled by means of a flap valve under control of a bimetallic thermostat strip located on the side of the exhaust manifold.

A great deal of attention has been devoted to obtaining good fuel economy on both engines and much development work has been undertaken in the direction of economical mixture strength to suit acceptable load and speed conditions. This, together with the attainment of optimum mixture distribution conditions, has resulted in very gratifying fuel consumption figures in terms of miles per gallon on the road.

The carburetor is developed to meter a widely variable range of mixtures according to load and speed requirements. Part-throttle mixtures are appreciably weaker than chemically correct. Part-load air to fuel ratios of 16.5 to 17.5/1 in the midload and speed range are obtained without difficulty. The carburetor is also equipped with an idling metering control and accelerator pump. The choke is manually operated and is interconnected with the throttle control for easy starting. Fuel is supplied to the carburetor by a diaphragm pump mounted on the crankcase and driven from the camshaft. The carburetor air intake is equipped with a resonator air intake silencer as standard, and for export use in dusty territories a combined silencer and oil bath air cleaner is available.

The design of the pistons and connecting rods follows proved practice. The forged-steel H-section connecting rod has a center length 1.78 times the stroke. Precision-fit white-metal steel-backed big end bearings are employed and the big end cap is secured by two 0.3125-in. high tensile steel bolts. A hollow piston pin runs in a bronze bush in the small end of the connecting rod and is fully floating in the aluminum-alloy piston. This pin is restrained, end-wise, by two flat-section Seegar circlips.

The piston carries two compression rings and one oil control ring, all above the piston-pin centerline. The compression rings have a width of 0.0937 in. and the oil control ring 0.1875 in. The top ring is rectangular in section whilst the second is L shaped or cornered, thus giving the effect of a tapered face ring for good oil control.

The coolant is circulated by means of an impeller pump mounted on the forward end of the cylinder head and driven by a triangulated V-belt drive from the forward end of the crankshaft. The water pump body is formed integrally with the thermostat housing and water outlet fitting to the radiator top tank. This arrangement lends itself to economical construction; it also makes it particularly easy to form a bypass between the cylinder-head outlet and the water pump inlet to ensure even and rapid warming of coolant within the engine before the thermostat opens.

The pump delivers the coolant to a shaped brass tube located within the cylinder-head water jacket and running its full length. This tube is ported to discharge coolant on to the spark-plug bosses and exhaust valve ports. Water circulates around the separate cast-iron cylinder barrels formed inte-

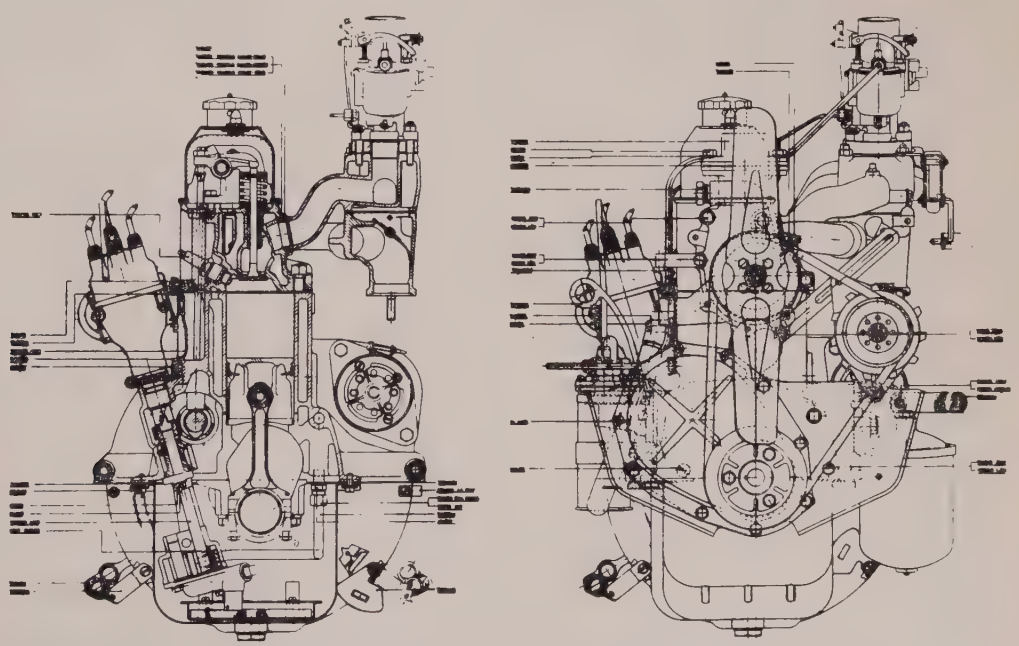

Fig. 30 – Vauxhall "square" engine

grally in the cast-iron cylinder block by thermosiphon action.

The 4-cyl engine (Fig. 31) gives 71 ft-lb torque, 117 bmep, 22 bhp, with a fuel consumption of 0.57 lb per bhp-hr at 1600 rpm, and 48 ft-lb torque, 80 bmep, 37 bhp, with a fuel consumption of 0.7 lb per bhp-hr at 4000 rpm.

The 6-cyl engine (Fig. 32) gives 104 ft-lb torque, 117 bmep, 27 bhp, with a fuel consumption of 0.58 lb per bhp-hr at 1400 rpm, and 78 ft-lb torque, 83 bmep, 58 bhp, with a fuel consumption of 0.65 lb per bhp-hr at 4000 rpm.

These performance figures are net figures for a complete engine with all standard accessories, including air silencer, generator, fan, standard ignition timing, and a complete vehicle exhaust system with silencer and tailpipe, using a 70-octane insensitive gasoline and corrected to standard barometer at 60 F carburetor air intake temperature.

In general the design of these engines is orthodox. In view of the stroke/bore ratio chosen and the general scantling dimensions, there is considerable latent development in the unit when better fuels are available and it is considered desirable to take more power out of the engines.

In conclusion, the author would like to point out that he has only been able to review briefly what he considers is a fair cross-section of British passenger-car engines in production today. As to the future it seems that the L-head engine is finished as a passenger-car engine, for reasons too obvious to state. It seems that the 4-stroke poppet-valve engine will predominate in the passenger-car-engine field for many years and in Britain it seems unlikely that the liquid-cooled engine will be replaced in quantity production.

The author wishes to thank his late employers, the Associated Ethyl Co., Ltd., London, and his present employers, Clifford Motor Components, Ltd., Birmingham, for permission to present this paper, and also to convey his grateful thanks to the following engineers and companies for much of the information given in the paper: W. M. Dunn, Alvis Ltd., Coventry; M. Cutler, Armstrong Siddeley Motors Ltd., Coventry; J. R. Rix, Austin Motor Co., Ltd., Birmingham; B. B. Winter, Humber Ltd., Coventry; D. Bastow, Jowett Cars Ltd., Bradford; W. M. Heymes, Jaguar Cars Ltd.,

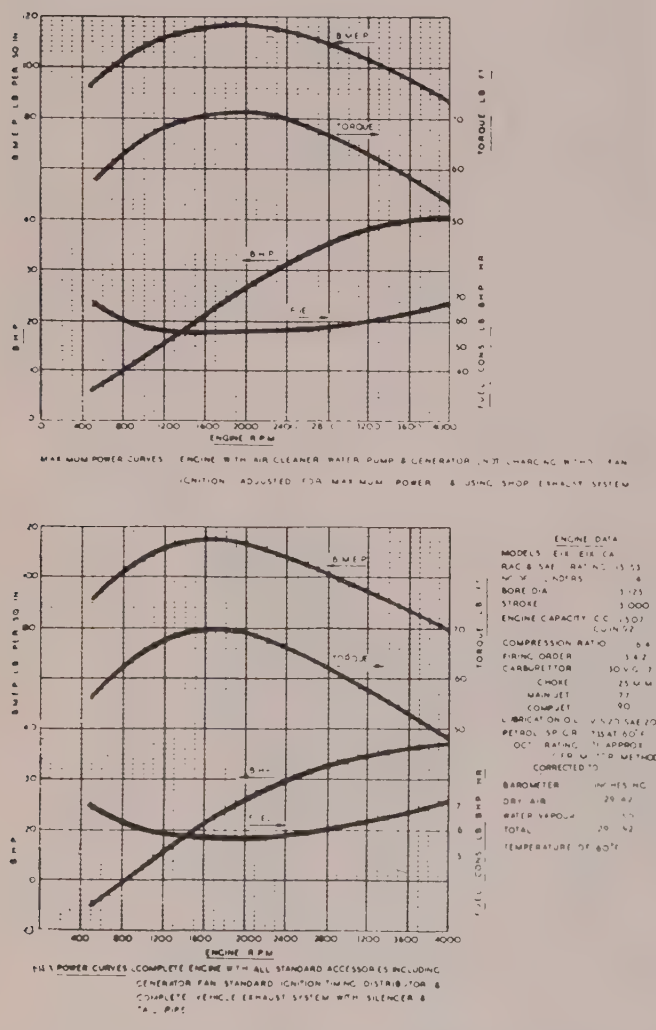

Fig. 31 – Performance curves for Vauxhall 4-cyl engine

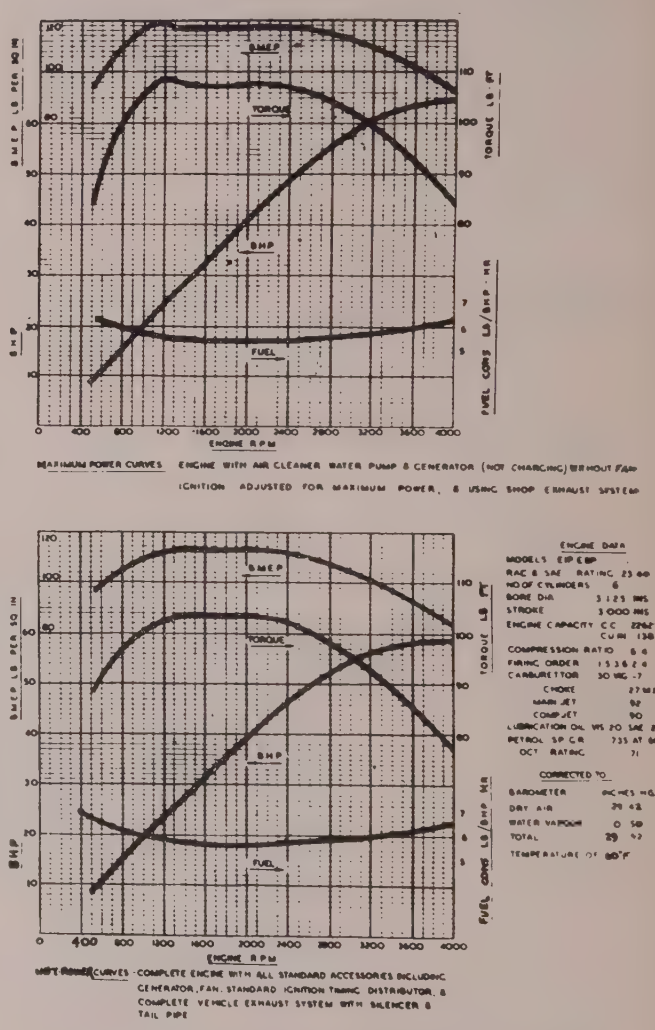

Fig. 32 – Performance curves for Vauxhall 6-cyl engine

Coventry; S. Enever, M.G. Car Co. Ltd., Abingdon-on-Thames; C. L. Jenner, Rolls-Royce Ltd., Derby; J. Swaine, Rover Co. Ltd., Birmingham; L. H. Dawtrey, Standard Motor Co., Coventry; and A. Larking, Vauxhall Motors Ltd., Luton.

DISCUSSION

Tells Why U. S. Engines Are the Way They Are

– John Dolza

General Motors Technical Center

AS soon as American automotive engineers were released from their war assignments, they outlined engine programs that were based on three ideas:

1. High-octane fuel will become available in the future.
2. Automobiles will have lower bodies.
3. New engines should incorporate all the economically sound features for higher efficiency, lower weight, and less bulk.

A great deal of analytical and test work in single-cylinder and multicylinder engines followed. The preliminary studies and tests proved that, as bore increased and stroke became shorter, the following benefits were achieved:

1. More room for valves, better breathing, and less pumping losses.
2. It became evident that the necessity and usefulness of the pentroof combustion chamber and inclined valves decrease and disappear when bore/stroke is large enough.
3. Shorter push-rods and stiffer valve mechanism eliminated the necessity of overhead camshaft.
4. Short stroke reduced piston and ring friction.
5. Greater overlap between crankpin and journal produced crankshaft suitable for high-compression operation.
6. Crankshaft rigidity improved bearing life.
7. Large-bore, short-stroke engines permit shortening the exhaust port and reduction of heat injection to the coolant.

Since American engines are of large displacement, to achieve it with large cylinder bores in in-line arrangement produced too long a powerplant, so the V was chosen.

Today, American V-8's are capable of operating at compression ratios up to 10 or 12/1, when the proper type of fuel comes along.

The V-8 engine has fewer bearings than its in-line counterpart of equivalent crankshaft rigidity. This factor also contributes to simplicity and low friction.

It is a very compact powerplant, which will give considerable latitude to the stylist or to hood height, width, and length. It is basically a lighter powerplant than the in-line.

It can be arranged to produce an even weight distribution between front and rear wheels more readily and with shorter wheelbase than its in-line counterpart.

Sees Little Originality In British Car Engines

– Maurice Olley

Chevrolet Motor Division, GMC

THE power output of British car engines is, on the whole, uninspiring. (See Table A.)

Table A – Power Output of British Car Engines

Make	Bhp/Cu In.
Alvis	0.46
	0.56
Armstrong Siddeley	0.572
	0.715
	0.95
Austin	0.57
Humber	0.46
Jowett	0.583
Jaguar	0.75
M.G.	0.60
	0.71
Rover	0.585
Vanguard	0.50
Vauxhall	0.40
	0.42

These horsepowers are quoted for 60 F air temperatures. Air at 100F would give even lower figures.

Probably the most important consideration is bhp per lb of engine weight. It would be interesting to have these figures. As pointed out by Dr. Lanchester many years ago, the small engine is at a definite disadvantage in this respect from dimensional considerations.

One cannot help wondering why the bold originality that is shown in the design of some British motorcycle engines is not seen in the passenger-car field. Notably lacking is the original thinking of the French or German designers, the aircooled flat engines, 2-stroke engines, miniature diesels, aluminum cylinders, and so on.

Almost the only engine showing originality in functional design for a small passenger car is the Jowett.

Contrast, for example, the Rolls-Royce Bentley design, which revolves around an engine produced with 4, 6, or 8 cylinders, with the Jowett design, which is based on the fundamental design of the vehicle.

One can add, perhaps, the Rover and Bentley F-head engines, and the Vanguard engine with four wet sleeves standing up unsupported in an open chamber forming the water jacket, which certainly has advantages in the foundry.

Continually, one finds British engineering thinking conditioned by British roads and climate. For instance, they have engines of small displacement designed for medium-speed performance, on the local roads, rather than for real high-speed performance on modern roads.

Or consider an engine designed and built without an oil filter, simply because there are no dust conditions in Britain.

Or again, consider the complication of design by the use of roller chains with tensioning and damping devices, simply because adequate supplies of silent chain are not available in Britain.

Which Will Better Fuel Bring: More Power or Better Economy?

– Errol J. Gay

Technical Consultant

THERE are a few points mentioned by Mr. Goodacre that I would like to re-emphasize. Fuel economy is a predominant factor in the design of British cars and engines. One quickly realizes this when a pound note ($2.82) is needed to pay for five imperial gallons of gasoline. There is also another important factor that the author touched on when mentioning the 14-ft garage length as determining the size of a car. British streets and garage facilities at private homes cannot store large cars for overnight parking, and their already congested city streets and parking facilities would be impossible for the daytime move-

ment of large cars. Thus, the smaller car and its engine become both a convenience and a necessity.

Mr. Goodacre has said that poor fuel quality in both the United Kingdom and many export markets limited British engine designers. He has noted the improved fuel in the U.K. starting in February, 1953. Recent events in the U.K., Europe, and many other overseas areas would indicate that a very considerable amount of competition between oil companies may result in improved fuel quality. It is estimated that by next year regular-grade fuels in the order of 76 Motor (80/81 Research) octane number, and premium-grade fuels of 80 Motor (88 Research) octane number will be available in many areas. These values are expected to improve even further in the years 1956-1958. Does the author feel that British manufacturers will move to take advantage of these improved fuels, and if so, will this move be toward greater power or increased fuel economy?

Describes English Ford Consul Engine

– V. G. Raviolo
Ford Motor Co.

IT is always interesting to review the developments in England. A study of the differences from U.S. practice and an analysis of the reasons for those differences may lead to a fresh perspective on our own problems. For example, less extreme climatic conditions have permitted the use of induction systems with less heat-transfer control than in the United States.

It is apparent that the use of less single-purpose machinery in England provides for greater design freedom, and one hopes that more advantage will be taken of this freedom in the future.

We of the Ford engineering staff in Dearborn are fortunate, in that we do a considerable part of the advance design studies for Ford's foreign operations, including Ford Cologne, Ford S.A.F. at Poissy, and Ford at Dagenham, England. Several years ago we laid down the bases for two new engines for England. One was to be for a 2350-lb car, and the other for a 2550-lb car. These became the Consul and Zephyr, respectively. The essential problem was to design a pair of engines, a four and a six, which could be produced in mix on one set of tools; that is to say, the two blocks were to go through the same transfer machine without changeover.

Fig. A shows transverse and longitudinal sections through the Consul engine. Both engines have a bore of 3.125 in. and a stroke of 3.000 in. This gives a displacement of 1508 cc for the four-cyl engine, and 2262 cc for the 6-cyl unit. The block is short and rigid, and the crankcase is quite deep. The original design was somewhat shorter, but it was necessary to elongate the block later, before production. The limitations on babbitt thickness in English production, at that time, made it imperative to have more bearing area than we would require here.

The valves are in-line, push-rod-operated. Special care was given to cam design to assure good operation at high engine speeds, as these engines operate in the gears more of the time than do U.S. cars.

Fig. B shows the near side of the Zephyr 6-cyl engine, and, except for the fan, which is quite small, the engine could pass as a conventional U.S. engine. You will note the cluster of distributor, and oil pump hidden below, fuel pump, and oil filter. This arrangement is similar to the U.S. Ford Six, and is used because of economies in machining and drilling.

You will note the duplex roller-chain drive for the camshaft. The crank is cast alloy steel. The pistons are autothermic aluminum, with steel struts for the control of thermal expansion. Two compression rings and one oil ring are used, and the wrist pin is clamped in the rod.

In the transverse section of Fig. A, the distributor and oil pump drive can be seen. Also, the valve spring (which in the drawing is right side up) is shown, as well as the manifold clamping arrangement. The section is cut deceptively at the head fastening. There are 10 fastenings for the Four, and 14 for the Six, equally spaced, and all are attached to the side walls rather than to the bores. These engines weigh 320 and 426 lb, respectively. The Four develops 47 bhp at 4400 rpm, and the Six, 65 bhp at 4000 rpm. Piston speeds at maximum horsepower are 2200 fpm and 2000 fpm. Maximum torque for the Four is 74 lb-ft at 2400, and for the Six, 112 lb-ft at 2000 rpm.

Fig. C shows the off side of the Consul 4-cyl engine. It is similar to the 6-cyl engine. Note especially the tubular steel exhaust pipe, which is clamped in place by the permanent-mold aluminum cast intake manifold. This clamp is proportioned to provide controlled manifold heat and has been quite successful. The wide flywheel housing mount can also be seen.

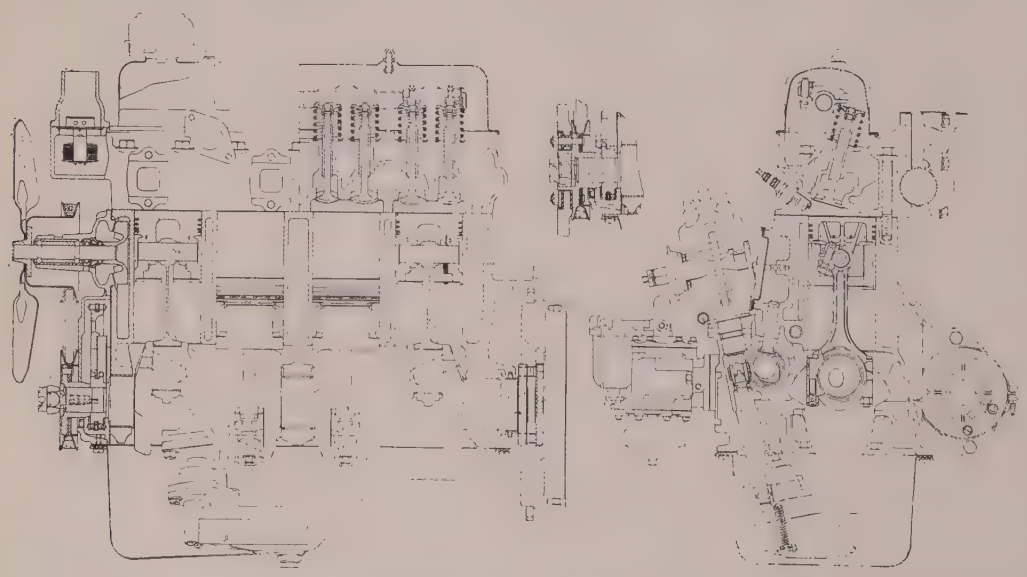

Fig. A – Ford Consul engine – bore and stroke: 3.125 x 3 in.; insert: torsional vibration damper of 6-cyl Zephyr engine

Fig. D is a close-up of the head and manifold assembly, showing the manifold arrangement in greater detail. Note the short, large-diameter valve springs. These springs are designed with damping coils (which can be plainly seen on the No. 2 exhaust spring at the center, which is assembled upside down).

Fig. E shows the block and head. The steel gasket can be seen lying on the block deck. From this, you can judge the relative size of the valves, the intake port diameter being 1.25 in., the exhaust, 1.0 in. The combustion chamber has a generous squish area, but a somewhat greater compromise for combustion softness was required for England, and the triangular shape was developed.

Points Out Some Differences Between British and U. S. Designs

– A. C. Sampietro
Willys Motors, Inc.

MR. Goodacre tells us the capacity of the various engines he describes, but the weight is quoted only for the Humber Blue Riband, stated to weigh 785 lb and to develop 116 bhp. Surely the weight of an engine for a given power is of greater practical importance than the piston displacement used to achieve the results, unless artificial formulas fix the amount to be paid in taxes, insurance, and the like. In England, while the license duty is now a flat rate, insurance premiums and in some cases even parking fees are fixed by the old RAC formula ratings.

Possibly, the influence of competition has created a fetish of power-per-liter in some sections, but why the bore and stroke of an engine, and not say its crankpin diameter, or any other combination of the many dimensions influencing power output, should still be taken as the main criterion of design merit seems strange.

None among the engines described by Mr. Goodacre is fitted with automatic valve-lash adjusters, perhaps our friends do not give the same importance that we do to freedom from service requirements, or their experience with hydraulic tappets may not have been too happy. We know that the oil fed to these units must be very clean, and in Great Britain the practice of treating cylinder blocks in the "kolene" bath or equivalent is not adopted, nor is oil filtration in the engine given quite the same attention it is here.

Some of the piston crowns shown in the illustrations would, I believe, be severely criticized by the good friend who has helped me with piston development. Could Mr. Goodacre give us some information regarding skirt clearance used by Rover and Jowett, and what sort of oil consumption do their engines show now?

I am particularly interested in the Armstrong Siddeley

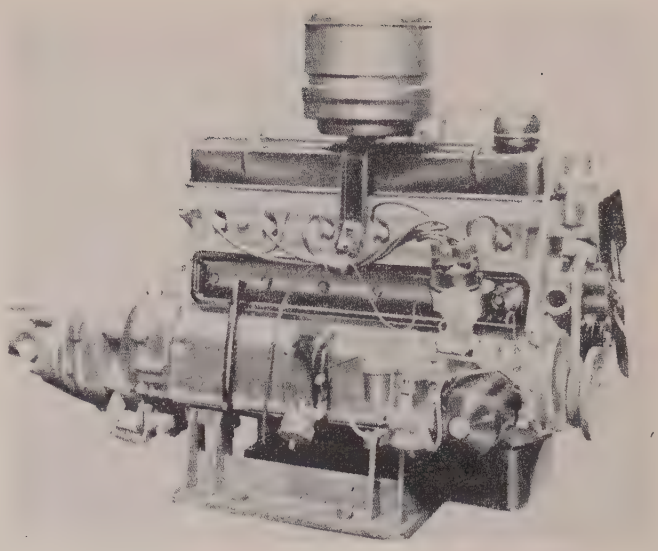

Fig. B – Ford Zephyr 6-cyl engine

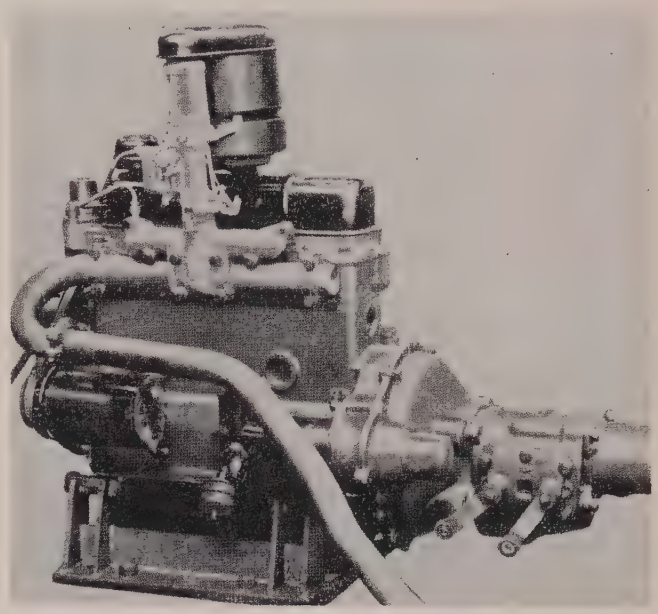

Fig. C – Ford Consul 4-cyl engine

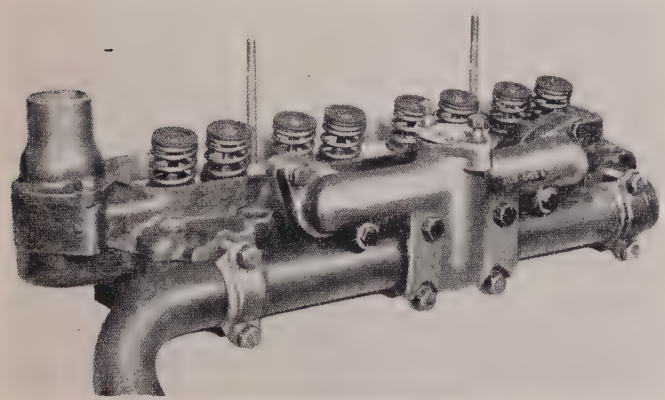

Fig. D – Inlet and exhaust manifolds of Consul engine

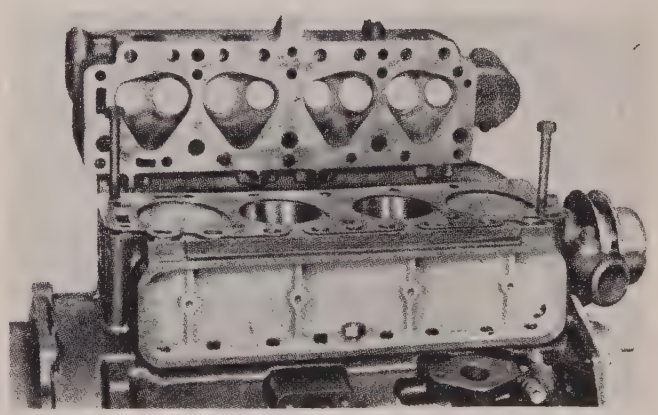

Fig. E – Cylinder block and combustion chambers

Sapphire, an engine that I have known more or less since its conception. In comparing its performance with the performance of the Jaguar, it would seem that up to, say, 5000 rpm, push-rod and rocker valve gear is not detrimental to performance. Various papers on the Chrysler engines and the excellent performance of the Riley and Bristol engines in England, and of the V.6 Lancia, tend to confirm this. When racing engine speeds (8000-10,000 rpm) are intended, direct operation of the valves is a must, but for an engine that may have to be serviced anywhere, valve clearances should be either automatically taken care of, or capable of being reset without having to remove two camshafts.

I wonder how much the shape of the ports in these two engines is dictated by the desire to create inlet turbulence and how much by other design necessities. In one of our experimental engines, two of the ports are rolled to the right, two to the left, and two straight, just to avoid studs and push-rods. There is no difference that we can find in the performance of the various cylinders.

This engine weighs 400 lb, it develops 121 bhp with 7.5/1 compression ratio, and a good deal more with 8.5/1, and the idle is low enough for use with automatic transmission. The displacement is 161 cu in. As I said before, it is only an experimental engine.

We still differ widely on our respective ideas of the minimum power that should be made available to the driver. Perhaps our hardest requirement is the acceleration we must have to pass our heavy trucks rolling along at 60 mph. A laden light car weighs about 3200 lb, and the British cars are not really any lighter, so a brisk acceleration, say 3 ft per sec per sec at 60 mph, takes 48 bhp to overcome net, rolling, wind resistance, and some transmission losses. This shows why we must have nearly 100 bhp in our light cars, and double that for a heavier, more-brilliant vehicle. Try commuting between Toledo and Detroit, and see what fun you can have with an underpowered car.

Nor is gas economy closely related to the engine power, weight, and shape of your automobile. Proper gear ratios and, unfortunately, the driver's foot have much more to do with it!

Author's Closure To Discussion

THE author agrees that the American postwar V-8 engine program outlined by Mr. Dolza seems to fit very well into the requirements of the North American Continent. The author is inclined to question the suitability of such powerplants for vehicles operated in less developed countries, when a simpler in-line powerplant might have considerable maintenance advantage. The author has the utmost admiration for the engineering of the postwar American V-8 engines that he has seen.

Mr. Olley's figures give an average power output per cubic inch of cylinder capacity for the British engines reviewed of 0.548 bhp per cu in. Figures published in the 1953 Ethyl Corp. American Passenger-Car Data Book give an average figure for 1953 American cars of 0.512 bhp per cu in., which, on the face of it, doesn't look too bad for the British engines, several of which are of prewar basic design. In regard to the omission of engine weights, the author apologizes for this, and agrees that the smaller the cylinder capacity the worse this figure can look. The author suggests that the British engines are less stereotyped in design than American passenger-car engines.

The author feels that original thinking in engine design is only to be applauded when this leads to commercial success.

In reply to Mr. Gay, the author is certain British engine builders will take full advantage of fuel quality gain, once this quality is stabilized in the major export areas in which British engine builders must trade. This stability does not seem to have been achieved yet. Some British engine builders are now offering alternative higher compression ratios for markets when fuel quality allows this.

The author has the greatest admiration for engine designs recently put into production by Mr. Raviolo's company, and feels certain that this view is held by many British engine designers.

In reply to Mr. Sampietro, the author agrees that push-rods and rockers need not be an embarrassment on passenger-car engines, when engine speed usually has to be kept below 5000 rpm for considerations of noise in the car. Again apologies for not including weight figures in this paper, but as the paper was long anyhow, much information available at the time of writing the paper had to be left out. The author is surprised that fixing engine size by cylinder capacity should cause any concern, as such assessment seems logical for any power or performance calculations. Established automatic valve lash adjusters seem to get into trouble from scale effect on small engines, further, the noise level of such engine is usually such that fixed tappets get by, although this state of affairs seems to be coming to an end in Europe. High-powered cars are very desirable to most people, if they can afford them and house them.

Preflame Reactions

discussion continued from page 618

systems are involved. If a cool flame front is present when the reaction heat is liberated, these methods are probably invalid.

Authors' Closure to Discussion

IT is obviously true that raw experimental data should ideally be converted directly to a final result with the use of computing machines as suggested by Professors Uyehara and Myers. In the present work the computing machine was used to obtain the final result after the raw data had been converted into a more convenient form. The suggestion concerning the use of the computer to obtain a minimum error curve from the raw data is good; however, in this case it would have involved far more computer storage capacity than was available.

The discussers' views concerning the effect of charge density upon the conductance would be well taken indeed, were it not for the fact that no discernible trend can be shown in the results presented in Fig. 5. The charge density for the nonreacting fuel-air mixture varied approximately 20% in these results. Surely this kind of change would be large enough to cause a discernible trend if this factor were the main reason for the lack of exact correlation shown in Fig. 5. The authors believe that, while certainly the density does affect the heat transfer conductance, to some degree the lack of exact correlation in this case can be largely attributed to experimental error.

The method presented here should be valid even though "the heat transferred is appreciable with respect to the energy released in preflame reactions." The maximum error in the cylinder volume incurred by assuming simple harmonic motion is 3.8% in the present work. This cannot be taken as a generalization, however, since most of the present work was done near top center, where the error is zero.

The authors concur with Mr. Rifkin that if preflame reactions do propagate in the same way as a normal flame the present treatment will not be valid.

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